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ELECTROLYTIC EFFECT OF VARIABLE CONCENTRATION, TEMPERATURE AND TURBULENCE ON DISSIMILAR METALS IMMERSED IN SEA WATER.

By GUYSBERT B. VROOM, LIEUTENANT COMMANDER, U.S.N., MEMBER.

ELECTROLYTIC ACTION.

The serious and costly results of electrolytic action on dissimilar metals, particularly in marine installations, are well known to designers and engineers. Piping, and heat transfer apparatus such as condensers and evaporators, containing salt water in various states of temperature, concentration and turbulence, often show rapid deterioration that can be charged directly to a potential difference between one or more dissimilar metals in contact with salt water, forming in effect a closed electrical circuit.

The effect, as is well known, is a rapid wastage and corrosive action, that appears finally as pin-hole leaks in piping and a general pitting of the surfaces in contact with the salt water, both in pipes and vessels.

It may well be that considerable electrolytic action takes place in boilers (since boiler feed water always contains sufficient salts and alkaline matter to render it an electrolyte) in some proportion depending on pressures and temperatures and, as indicated by the experimental data about to be described, the turbulence of the water in a steaming boiler.

The writer has observed, in the course of some years' experience, the highly potent effects of electrolytic action, ranging from complete failure of copper injection and overboard discharge pipes within three months, to the slow pitting of composition valves and piping over much longer periods.

Curiosity as to the variation of potential difference (with resultant current flow and electrolytic action) led to the following experiments. These covered a period of three months and were conducted as carefully as conditions permitted, in an improvised laboratory on the U.S.S. *Lexington.*

The object of the investigation was to determine approximately

(*a*) Voltage drop between two dissimilar metals immersed in sea water.

(*b*) Rate of change of voltage drop with change of temperature of electrolyte (sea water).

(*c*) Rate of change of voltage drop with change of temperature of electrolyte (sea water + NaCl to form a saturated solution).

(*d*) Rate of change of voltage drop with rate of change of *turbulence* of electrolyte at various temperatures.

In order to avoid the introduction of variable factors, so that more exact comparison of results might be made, the same equipment was used for all series of experiments, including the anodic and cathodic metals.

The set-up consisted of a glass jar (illustrated in Figure I) with a Bakelite cover. The cover carried vertical cylindrical Bakelite rods which held the anode and cathode, plates of mild steel and

Navy composition "G". To the latter were soldered lead out wires, insulated from the electrolyte, these leads going directly to a millivoltmeter. A small insulated immersion heater was used to maintain constant temperatures of the electrolyte. Turbulence was obtained by a small impeller immersed deep in the electrolyte and driven by a fan motor controlled by a rheostat which permitted close impeller speed control. This was necessary to obtain comparable results, since the turbulence was purely relative, and therefore the voltage drop-turbulence comparisons had to be on the basis of "Rest," "Low," "Medium," and "High," where "High" was maximum impeller speed, "Medium" was half speed, and "Low" quarter speed.

As noted above, all surfaces except the test plates of dissimilar metals, were carefully insulated from the electrolyte.

As shown in the notation accompanying Figure I the immersed test plates were 1 by 2.5 inches, held parallel to each other 4 inches apart.

The "normal" sea water used was obtained about 100 miles off shore in the Bay of Panama, as drawn from the ship's salt water supply system. No chemical analysis was made of the water.

The "saturated" solution was made up of the normal sea water + NaCl (table salt).

Proceeding now with a description of the results obtained from a number of duplicated and checked readings, Figure II shows four faired curves, of which Series I and II show the potential difference between the plates as recorded under identical conditions, viz.:

(*a*) Range of temperature 90 degrees to 128 degrees F.
(*b*) Electrolyte at Rest.
(*c*) "Normal" sea water.
(Series II is a check series against Series I.)

Series III and IV of Figure II show the potential difference under identical conditions, viz.:

(*a*) Range of temperature 90 degrees to 128 degrees F.
(*b*) Electrolyte at Rest.
(*c*) Saturated saline solution.
(Series IV is a check series against Series III.)

Figure III is a family of curves; each curve of the family shows the potential difference when the temperature of the electrolyte is held constant, but the turbulence is varied. The relative values chosen for "turbulence" are explained above. Figure III shows what is here termed "Descending" turbulence for lack of better descriptive phraseology; *i.e.*, the readings were taken commencing with maximum turbulence; the impeller was then slowed to half speed, and the reading noted when the potential difference became constant; then a similar reading was taken at quarter impeller speed, and the final reading taken when the elctrolyte had come to rest.

Figure IV is a family of curves similar to Figure III, the readings in this case, however, are for "Ascending" turbulence, *i.e.*, starting with the electrolyte at rest, to maximum turbulence.

GENERAL COMMENT.

The experimental data shown in Figures II, III and IV, exhibiting the potential difference between two selected metals ordinarily used in equipment and machinery units subjected to or containing sea water at different temperatures and at different degrees of concentration and turbulence, would seem to indicate that the resultant electrolytic action, and hence wastage, pitting and general deterioration, takes place in mathematical relation to the amount of variation of temperature, turbulence and concentration, and/or a combination of these variables.

It would seem to be reasonable to expect that a boiler at high temperature (and, of course, high rate of turbulence of the electrolyte [boiler water]) would show pitting and "corrosion" that could be directly traced to electrolytic action. It would follow that the "corrosion" and pitting would be severest where the temperatures were highest and where the turbulence due to circulation and evolution of steam was greatest. It would seem to follow from the above that the electrolytic action would vary throughout the boiler, perhaps widely within very small areas, so much so as to defeat any efforts to apply counteractive E.M.F., which has been proposed from time to time.

This data appears to show, as far as it goes, that definite and varying amounts of "corrosion" and pitting may be expected in

salt water piping and vessels containing salt water, according to the materials of which they are made and, in the same equipment according to the relative " forcing" which they undergo.

For example, condenser tubes might be expected to fail sooner at high rates, and in tropical waters, than those in service at the same rates but in temperate waters; and so on for all similar equipment.

It would be a matter of considerable interest to elaborate the above simple and elementary experiments to include combinations of metals ordinarily used in marine boilers and machinery, at much higher temperatures than those used in these series; and to supplement the electrolytic determinations thus made with metallurgical examinations to determine, for instance, the tensile strength of metals after given periods of subjection to electrolytic action. Such information might materially assist in improving commonly used alloys, and permit their arrangement in orders and classes best suited to applications where the approximate conditions, conducive to electrolytic effects, were known or could be determined relatively to standards established in a laboratory.

The above described qualitative examination of the variation of potential difference—and resultant current flow—between metals immersed in sea water indicates that as temperature and turbulence, either or both, increases, a wastage caused by electrolytic action may be expected that has some approximate relation to temperature and relative turbulence.

Only one alloy was used in the above qualitative determinations, and it is therefore proposed to examine quantitative results by subjecting ten different specimens of non-ferrous alloys to a 60-day immersion test, using the set-up described above, in which the alloy under test will be connected by a metallic path to a similar plate of mild steel. A comparison of final weight with the original weight, together with microscopic examination of the surface of the specimen and a chemical analysis of the final electrolyte will, it is hoped, yield some clues as to the probable life of the several alloys commonly employed in salt water service.

The first series of tests will therefore be made at a temperature of 100 degrees F., with no turbulence of electrolyte; to be

followed by series of tests at more elevated temperatures, with and without turbulence.

The results are to be published as completed from time to time.

The ten specimens selected have the following characteristics:

CHARACTERISTICS OF SPECIMENS.

	No. 4	No. 5	No. 6 (Navy M)	No. 7 (Navy Sc.)
Sn	4.25 Min.	6.00 Min.	6.25 Min.–7.25 Max.	4.00 Min.
Pb	5.00 Max.	2.00 Max.	1.0 Min.–2.0 Max.	3.00 Max.
Cu	83.00 Min.	86.00 Min.	86. Min.–91. Max.	80.–83.
Fe	....	....	.25 Max.	.20 Max.
Zn	Rem.	Rem.	1.5 Min.–5 0 Max.	Rem.

	No. 9	No. 22	No. 24	No. 25	No. 32½ White Metal (Plumbing)	Forging Brass
Sn	.50	10.00	12.00	10.00		.35
Pb	.60	....	....	....		1.5–2.5
Cu	57.	88.00	88.00	90.00		58.–61.5
Fe	1.50	....	....	....		1.5
Zn	40.	2.00	....	....		Rem.
Mn	.02	....	....	....		....
Al	.40	....	....	....		....

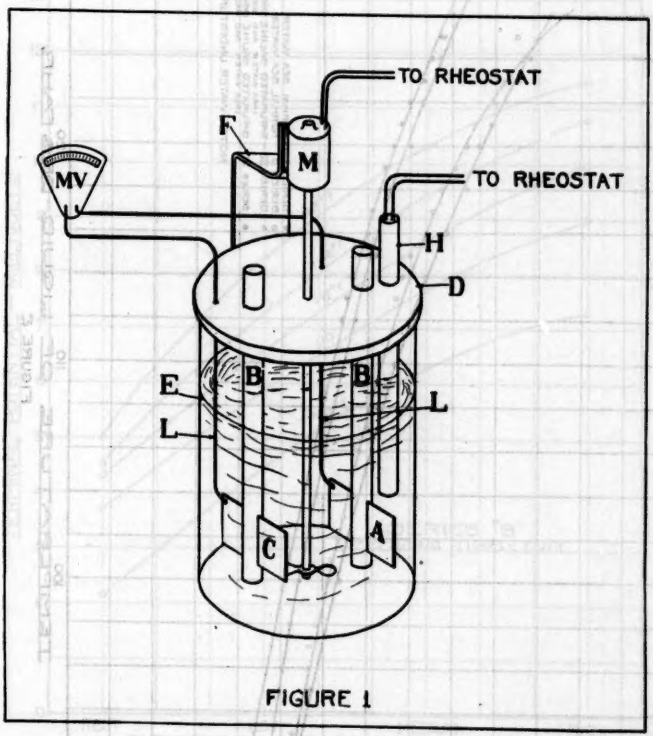


FIGURE 1

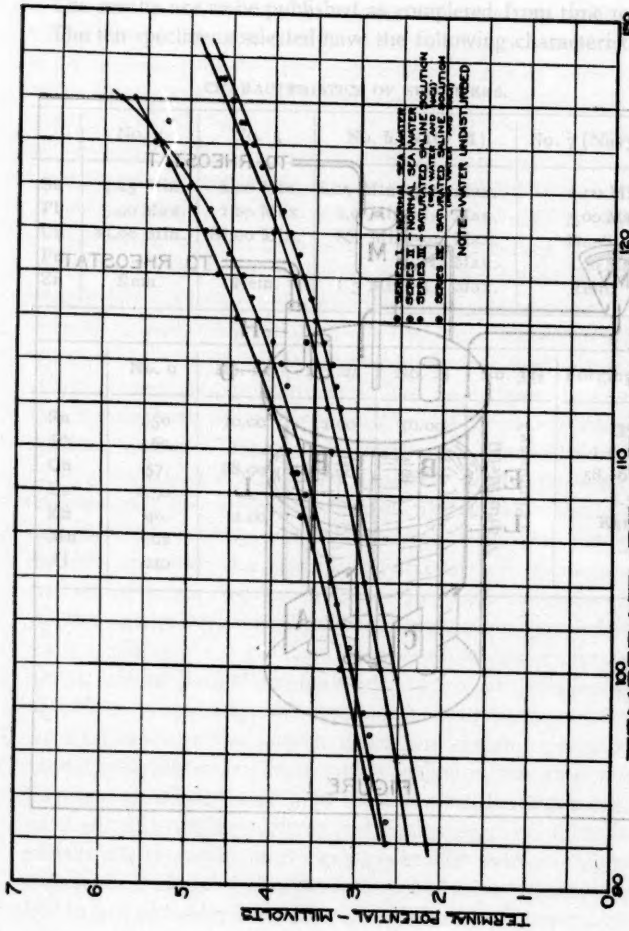

SERIES I = NORMAL SEA WATER
SERIES II = NORMAL SEA WATER + SATURATED SOLUTION (NaCl)
SERIES III = SATURATED SALINE (NaCl) SOLUTION
SERIES IV = SATURATED SALINE (NaCl) SOLUTION
NOTE - WATER UNDISTURBED
TERMINAL POTENTIAL - MILLIVOLTS
0
1
2
3
4
5
6
7
80
100
110
120
130
TEMPERATURE OF LIQUID - DEG. FAHR.

FIGURE 2

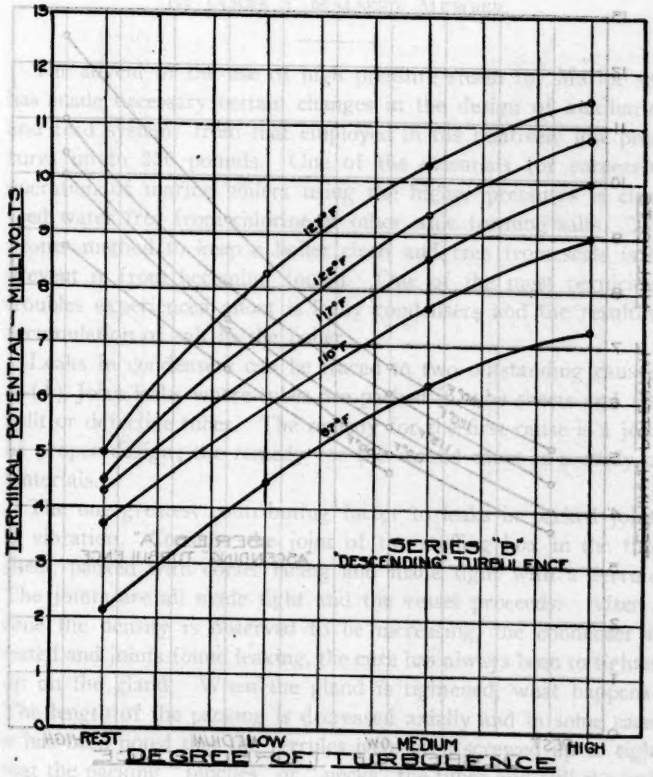

TERMINAL POTENTIAL — MILLIVOLTS
0
1
2
3
4
5
6
7
8
9
10
11
12
13
127°F
122°F
117°F
110°F
97°F
SERIES "B"
"DESCENDING" TURBULENCE
REST
LOW
MEDIUM
HIGH
DEGREE OF TURBULENCE

FIGURE 3

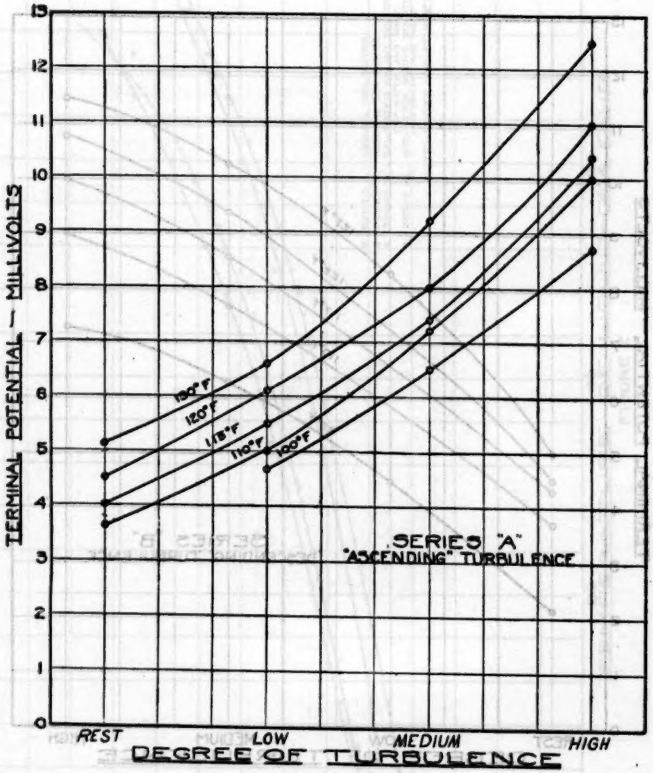

TERMINAL POTENTIAL — MILLIVOLTS
0
1
2
3
4
5
6
7
8
9
10
11
12
13
REST
LOW
MEDIUM
HIGH
DEGREE OF TURBULENCE
SERIES "A"
"ASCENDING" TURBULENCE
130°F
120°F
115°F
110°F
100°F

FIGURE 4

CONDENSER EXPERIMENT—S.S. *BAKERSFIELD.*

By JAMES S. MALSEED, MEMBER.

The advent of the use of high pressure steam for Marine use has made necessary certain changes in the design of auxiliaries and feed systems from that employed in the relatively low pressures up to 250 pounds. One of the essentials for successful operation of marine boilers using the higher pressures is clean feed water free from chlorine or other scale forming salts. The proper method to keep a boiler clean and free from scale is to prevent it from becoming fouled. One of the most pernicious troubles experienced afloat is leaky condensers and the resulting accumulation of salt in the boiler.

Leaks in condensers can be traced to two outstanding causes: —(1) Joint leaks where tubes are packed in tube sheets and (2) split or defective tubes. The remedy for the first cause is a joint of proper design; the remedy for the second cause is a study of materials.

The one greatest contributing factor to leaks in packed joints is vibration. Consider the joint of the stuffing box in the tube sheet, packed with corset lacing and made tight with a ferrule. The joints are all made tight and the vessel proceeds. After a time the density is observed to be increasing, the condenser is tested and joints found leaking, the cure has always been to tighten up on the gland. When the gland is tightened, what happens? The length of the packing is decreased axially and in some cases it has been noted that the ferrules have been screwed up so tight that the packing "pinches" or "necks" the tubes and still the leak continues or grows worse.

Picture the action of the tubes in vibration—Figure 1. The vibration tends to rock the tube over the packing and when the gland is tightened, the packing decreases in length and the shorter the length of packing the more readily the tube rocks over it. In 90 per cent of the cases it is found that the support plate is placed

equi distant from the tube sheets, this also allows vibration to continue without the deadening effect that can be accomplished by a sufficient number of support plates, so spaced that the nodes of vibration are broken and the effect of vibration deadened.

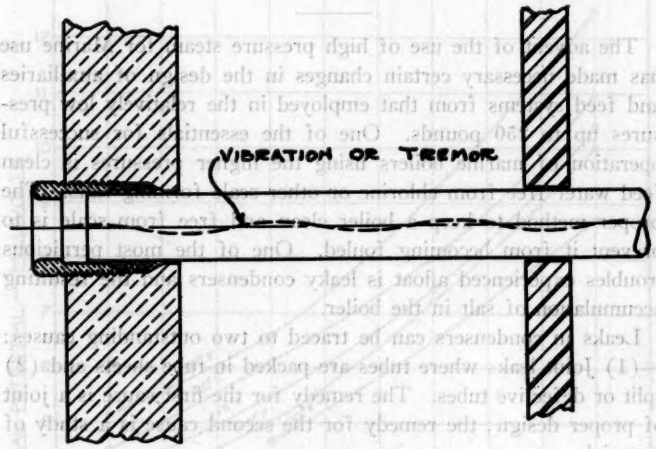


FIGURE 1.

Tubes rolled into tube sheets, when properly expanded, form the best joint to maintain tightness, as the tube is made tight with the sheet for the full length of the hole in the sheet.

Condensers with tubes expanded into sheets at one end and packed at the other end have been used extensively with good results and tubes expanded at both ends have been used with the tubes bowed in either the horizontal or vertical direction, to allow for expansion.

The use of straight tubes expanded into sheets at both ends with no provision for expansion had not been used at sea, though considered by several prominent marine designers to be a feasible method, as the amount of expansion was not considered sufficient but that the elasticity of the metal would accommodate it. The Shipping Board decided to make such an experiment so that

the information from an actual trial would be available to the Merchant Marine and appropriated the necessary funds from the Engineering Development Program to carry out the experiment.

The S.S. *Bakersfield* was selected for the trial installation because of its propelling machinery being reciprocating engines with scotch boilers. This selection was made for two reasons:—(1) In case of tube joint failure and consequent salting of the boilers the scotch boilers would be better able to withstand the water condition than a water tube type and (2) the reciprocating engine operating at relatively low vacua would create a higher temperature difference between the sheet and the tubes than would a turbine installation, and would be therefore, the maximum condition for strain on the joint of the tube in the sheet.

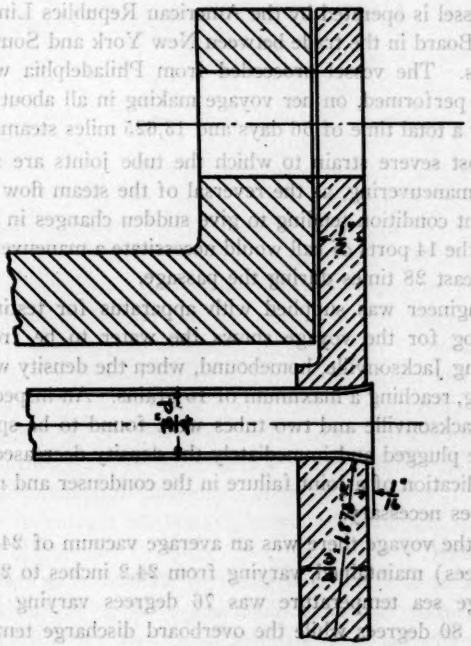


FIGURE 2.

The work of installation was completed under the direction of Capt. J. E. Bailey, Director, M. & R. Division of the Merchant Fleet Corporation and consisted in removing the tubes and tube sheets, replacing them with new tube sheets of Muntz Metal and tubes of Shipping Board specification, Admiralty Metal. The joint is shown in Figure 2.

The condenser is of the two pass type having 3500 square feet of cooling surface with a cast iron shell and ¾ inch diameter tubes, the circulating water entering at the bottom and discharging at the top. No change from the original design was made in tube layout, but the tube sheet was decreased in thickness from 1 inch to ¾ inch and faced down in way of the flange to ½ inch thickness. Upon completion the condenser was tested with water for tightness and the vessel then proceeded.

This vessel is operated by the American Republics Line for the Shipping Board in the trade between New York and South American Ports. The vessel proceeded from Philadelphia where the work was performed, on her voyage making in all about 14 ports of call for a total time of 56 days and 13,625 miles steaming.

The most severe strain to which the tube joints are subjected is during maneuvering, as the reversal of the steam flow gives an intermittent condition tending to give sudden changes in temperature. So the 14 ports of call would necessitate a maneuvering condition at least 28 times during the passage.

The Engineer was supplied with apparatus for testing water and the log for the voyage shows the water to be fresh until approaching Jacksonville, homebound, when the density was noted to be rising, reaching a maximum of 15 grains. An inspection was made at Jacksonville and two tubes were found to be split, these tubes were plugged and immediately the density decreased. There was no indication of a joint failure in the condenser and no rerolling of tubes necessary.

During the voyage there was an average vacuum of 24.3 inches (139 degrees) maintained, varying from 24.2 inches to 25 inches, the average sea temperature was 76 degrees varying from 60 degrees to 80 degrees while the overboard discharge temperature

averaged 102 degrees which would indicate a temperature difference of approximately 40 degrees at top of the condenser, between the entering steam and the overboard discharge.

The results so far obtained in this experiment indicate that no trouble will be experienced with condenser construction of this nature and will eliminate packing leaks, thereby decreasing maintenance and labor, not only on condensers but the boilers and the whole system affected by salt leaks.

THE BALANCING OF ROTATING MASSES.

By LIEUTENANT H. G. EBERHART, U. S. NAVY, MEMBER.

It is believed that the question of the balancing of rotating masses is one that is not thoroughly understood by the majority of operating engineers. The art of balancing is one that is comparatively new and by many is incorrectly assumed as being too complicated to be understood by any one but the "expert." Others, while not understanding its underlying principles, know that the art exists and conclude that if balancing is necessary in some cases, it must be necessary in all cases. The result is that on one hand unbalance is ignored, on the other it is assumed to be the evil influence that is the cause of all real or imaginary vibration. When this latter viewpoint is taken work requests are submitted to balance this, that, or what not, with no real understanding of the necessity therefore, of its relative cost nor of the fact that except for handling and rigging charges, it costs as much to balance a small object as a large one.

Without understanding the underlying principles of, and the procedure employed to accomplish this operation, the engineer is at the mercy of the balancing machine operator as to the costs incurred and results obtained just as the owners are at the mercy of the engineer who wants every object balanced to an extreme degree of precision. The following exposition is written with the view of acquainting those who are not already conversant with the subject, with the principles underlying the balancing of rotating masses, the necessity therefore, and the manner in which it is accomplished.

Any mass, as it is moved, either by translation or rotation, may impress certain harmonic disturbances on its supports. In the case of a rotating mass attached to a shaft, the disturbance may be manifested as transverse vibration, torsional vibration, or both.

In systems consisting of rotating masses only, transverse vibration is caused by an unbalanced condition of these masses. The correction consists of applying the necessary weights to compensate for this unbalance. This operation is called balancing. Since it is performed only on systems already in existence, torsional vibration, despite its importance, will be neglected and the following discussion will be based on the assumption that the bodies considered are absolutely rigid.

A mass exactly symmetrical with reference to its axis of rotation will, except for torsional disturbances, rotate without vibration. Such a mass would be in balance. It is, however, difficult to satisfy this condition, since, in the process of manufacture, some irregularities in mass distribution are nearly always present and unbalance to some degree exists.

Unbalance is of two kinds, static and dynamic. They are alike in their effect, *i.e.*, either produces vibration when the mass is rotated. Their difference lies in the cause and manner of correcting this defect. It is further to be noted that a mass may be in static balance and yet, due to dynamic unbalance, vibrate badly when rotated.

As previously stated, most bodies, as manufactured, are in either static or dynamic, or both static and dynamic, unbalance. In order to explain these various conditions, a rotor, shown as Figure 1, will be considered.

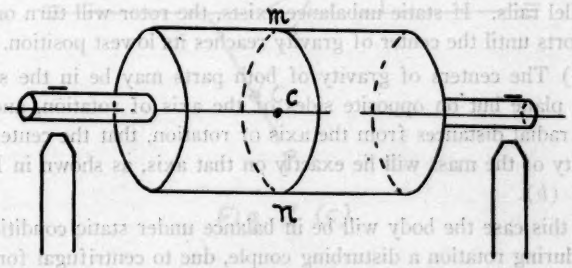


FIG. 1

For the purpose of discussion, imagine this body to be divided into two parts by m n. Each section will have its own center of gravity. The center of gravity of the mass will be the resultant of the centers of gravity of all sections comprising it. In such, or any similar system, the following typical cases of unbalance may arise, *viz:*—

(*a*) The centers of gravity of both parts may be in the same radial plane and on the same side of the axis of rotation, as shown in Figure 1 (*a*).

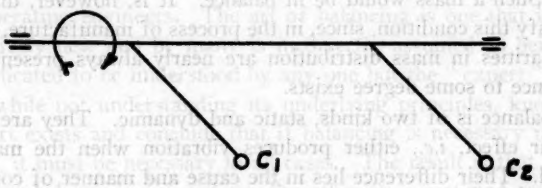


FIG. 1 (a)

The center of gravity of the mass will be in the same plane, at a definite distance from the axis of rotation. This condition, since it can be detected by static test, is called one of static unbalance. It can be found by supporting the shaft on absolutely horizontal, parallel rails. If static unbalance exists, the rotor will turn on its supports until the center of gravity reaches its lowest position.

(*b*) The centers of gravity of both parts may be in the same axial plane but on opposite sides of the axis of rotation, and at such radial distances from the axis of rotation, that the center of gravity of the mass will lie exactly on that axis, as shown in Figure 1 (*b*).

In this case the body will be in balance under static conditions, but, during rotation a disturbing couple, due to centrifugal forces, will act on the rotor. This couple rotates with the mass and tends to make it oscillate thereby producing vibration in the foundation. Such a case, is one of dynamic unbalance.

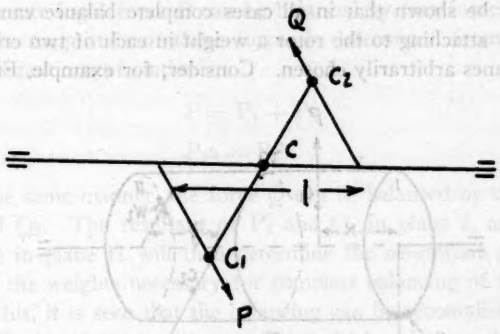


Fig. 1 (b)

(*c*) Most generally, the centers of gravity, C_1 and C_2, lie on different radii, in such a manner that the center of gravity of the mass does not lie in the axis of rotation. This is shown in Figure 1 (*c*).

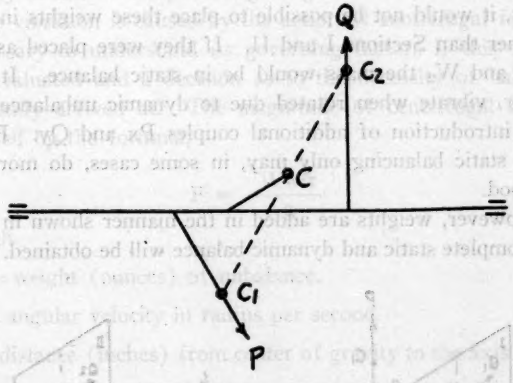


Fig. 1 (c)

During rotation a system of two forces formed by the centrifugal forces, P and Q, will act on the body. This system of forces can always be reduced to a couple acting in an axial plane, and a radial force. Static and dynamic unbalance occur together.

It will be shown that in all cases complete balance can be obtained by attaching to the rotor a weight in each of two cross sectional planes arbitrarily chosen. Consider, for example, Figure 2.

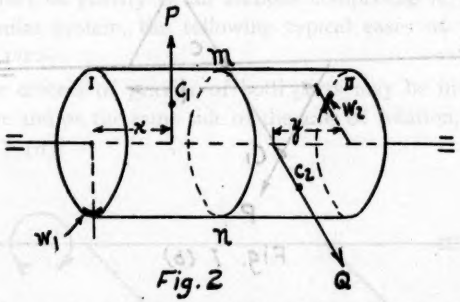


Fig. 2

Due to unbalance, two centrifugal forces, P and Q, act on the rotor during motion. The static unbalance can be corrected by the application of single weights placed in the same axial plane but opposite to the unbalance to be corrected. For mechanical reasons, it would not be possible to place these weights in a location other than Sections I and II. If they were placed as shown by W_1 and W_2 the mass would be in static balance. It would, however, vibrate when rotated due to dynamic unbalance caused by the introduction of additional couples Px and Qy. For this reason, static balancing only may, in some cases, do more harm than good.

If, however, weights are added in the manner shown in Figure 2 (*a*) complete static and dynamic balance will be obtained.

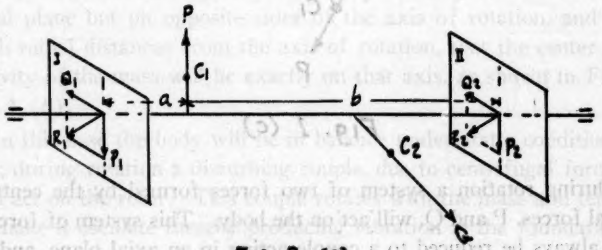


Fig 2 (a)

The centrifugal force P can be balanced by two forces P_1 and P_2. The magnitude of these forces is determined from the following equations of statics.

$$P = P_1 + P_2$$

$$P_1 a = P_2 b$$

In the same manner, the force Q can be balanced by the forces Q_1 and Q_2. The resultant of P_1 and Q_1 in plane I, and of P_2 and Q_2 in plane II will then determine the magnitude and position of the weights necessary for complete balancing of the rotor. From this, it is seen that the balancing can be accomplished without difficulty if the position and magnitude of the unbalance is known. For determining this unbalance various kinds of balancing machines are used. In general, these are of two classes;—

(*a*) Lawaczeck-Heymann—one in which the location and magnitude of unbalance is determined from records of the vibrations.

(*b*) Akimoff—one in which the same information is determined by means of compensating weights.

Since vibration is caused by the action of centrifugal forces, it is necessary to understand its governing factors if this force is to be evaluated and a decision as to the necessity of balancing, intelligently arrived at. The magnitude of centrifugal force is expressed by the formula,

$$F = \frac{Ww^2r}{g}$$

in which

W = weight (ounces) of unbalance.

w = angular velocity in radius per second.

r = distance (inches) from center of gravity to the axis of rotation.

g = acceleration due to gravity.

In the case of simple static unbalance these are the only factors causing vibration. This is the condition usually found in a relatively thin rotor or disc, shown as Figure 3.

It is seen from this Figure that the center of gravity must lie within relatively small axial limits, consequently any couples existing must be small as compared to the centrifugal force acting on the shaft. For this reason the unbalance will be largely static. In a mass of this kind, due to its symmetry, W and r are likely to be small, so w is the controlling factor in its unbalance.

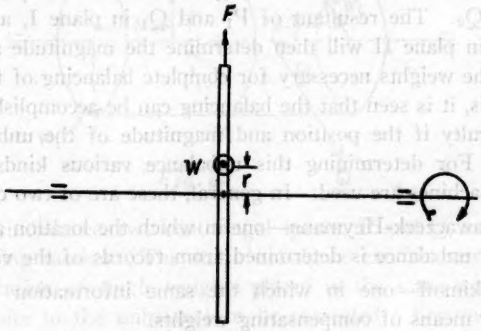


Fig. 3

In the case of dynamic unbalance only, no disturbing force is exerted by the center of gravity of the mass since it lies on the axis of rotation as shown in Figure 1 (*b*). Such unbalance as exists is due to the presence of a couple caused by forces P and Q. This couple tends to cause the ends of the shaft to travel in a circular path. Since it is constrained by bearings, it oscillates and vibration results. Its magnitude is given by the formula,

$$C = \frac{Ww^2rl}{g},$$

in which

W, w, r and g are as given previously

l = perpendicular distance between the lines of action of the forces (inches).

In the case of masses of relatively large axial dimensions, dynamic unbalance is probable since it is possible for l to be large,

even though the object is in static balance. For this reason static balance, only, is often ineffectual in the case of rotating masses.

From a consideration of the two formulae previously given it is seen that the amount of unbalance varies directly with W, r and l, and as the square of w. The effect of these variables is as follows, *viz:*—

(*a*) The higher the speed of rotation the greater the necessity for balancing.

(*b*) The shorter the axial length in proportion to the diameter, the less the need for dynamic balancing and, conversely, as the proportion of axial length to diameter increases, the greater becomes the necessity for dynamic balance.

(*c*) In a symmetrical, homogeneous body, W and r should be small, but the effect of a given unbalanced weight, W, decreases as the proportion of the mass near the center to the total mass increases.

The following cases will illustrate these points.

CASE I

NARROW ROTOR

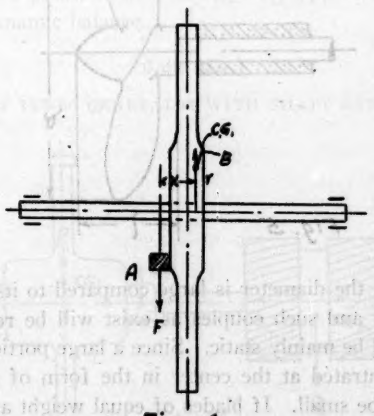


Fig. 4

Assume the center of gravity to be as shown. It is the resultant of the centers of gravity of any number of sections. Due to the configuration and symmetry of the Figure it must lie within comparatively narrow axial limits, hence any couples set up will be relatively small. In the Figure as shown, static unbalance exists and if smooth running is to be obtained, it must be removed. Unless this is a high speed wheel, dynamic unbalance is not indicated.

The ordinary static test, balancing on stationary rolls, will not show the axial location of the center of gravity. The static unbalance can be corrected by adding weight on the opposite side of the axis of rotation. If it is added at A, the static unbalance will be removed but dynamic unbalance will be increased by introducing a couple FX. It is better to remove weight from the heavy side of the disc at B by boring. This avoids the possibility of increasing the dynamic unbalance.

CASE II.

PROPELLER.

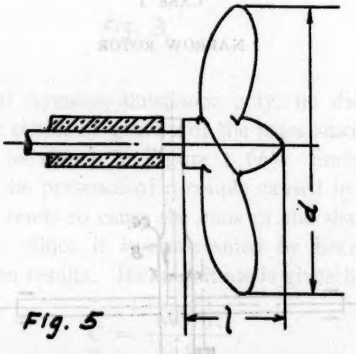


Fig. 5

In this case, the diameter is large compared to its length, hence l will be small and such couples as exist will be relatively small. Unbalance will be mainly static. Since a large portion of the total mass is concentrated at the center in the form of a symmetrical hub, r should be small. If blades of equal weight are selected, W should be small and so the speed of rotation becomes the controlling factor. If this is such that the introduction of weight

to correct static unbalance introduces a large couple despite the small value of l, dynamic unbalance will manifest itself and must be corrected. It is believed, except in the case of very high speed, light weight propellers, where w^2 assumes considerable proportions relative to the mass of the propeller, that dynamic balancing is a needless refinement.

In considering the action of a propeller, it should be borne in mind that when revolving in air, the conditions are not those encountered in service. When submerged, the propeller is working against a relatively resistant medium. The pressure on the bottom blade is greater than that on the top blade because of its deeper submergence. Each small section of the blade has a different thrust because of slight variations in pitch. The entire propeller is hung on the end of a long shaft, supported at one end only, and receives more or less periodic impulses from the source of power. From a consideration of these factors, it is believed that the vibration existing in a propeller shaft, is largely of the torsional rather than the transverse type, provided the propeller is in static balance. It is common knowledge that a destroyer propeller will vibrate badly if a blade is bent even though no metal is lost. The vibration in this case is caused by a change in effective area or pitch, rather than by any appreciable change in the static or dynamic balance.

CASE III.

ROTOR OF TURBO GENERATOR WITH SHAFT AND GEARS.

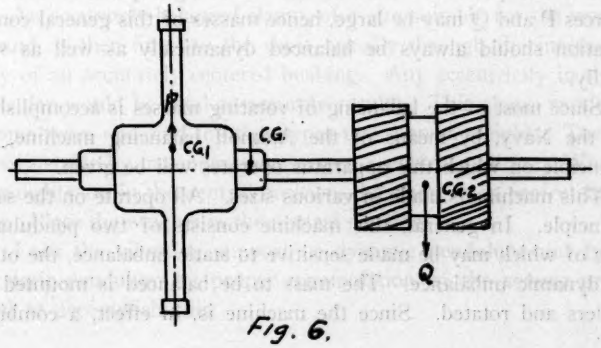


Fig. 6.

The disc alone having short axial length will have very little, if any, dynamic unbalance. But since the rotor is but a part of the system the center of gravity of the mass will fall, probably at CG. Due to the length of the mass, relative to its diameter, dynamic unbalance will very likely exist. Such a unit should be balanced dynamically.

CASE IV.

LONG ROTOR, ARMATURE, OR GEAR.

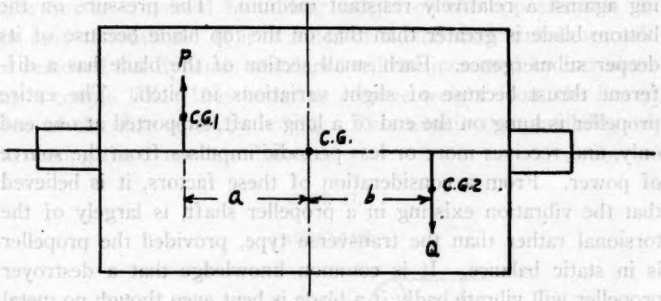


Fig. 7

In this case, it is possible and probable, that dynamic unbalance will exist even though the mass be in static balance. Due to the length of the moment arms, a and b, couples of considerable magnitude are possible. Due to the usual high speed of rotation, forces P and Q may be large, hence masses of this general configuration should always be balanced dynamically as well as statically.

Since most of the balancing of rotating masses is accomplished, in the Navy, by means of the Akimoff balancing machine, the principle on which this apparatus operates will be given.

This machine is made in various sizes. All operate on the same principle. In general, this machine consists of two pendulums, one of which may be made sensitive to static unbalance, the other to dynamic unbalance. The mass to be balanced is mounted on rollers and rotated. Since the machine is, in effect, a combina-

tion of two pendulums, unbalance will be indicated by vibration in the machine. By means of an adjustable clamp, which carries compensating weights, the proper compensation is made so that no vibration occurs as the mass is rotated. With the data thus obtained, the amount and location of the weight to be added to put the object in balance is determined.

The procedure, by which this information is obtained, follows two general lines, *viz;*

(*a*) Obtain and correct for static and dynamic unbalance separately, or,

(*b*) Obtain the amount of static and dynamic unbalance in one operation and correct both at the same time by applying weights, equilibrating the resultant of these forces.

Procedure (*a*), since it is a longer process, is somewhat out of date. While it arrives at perfectly correct results, the extra work involved renders it the less favored method of balancing. It is believed, however, that an understanding of (*a*) will aid in the handling of (*b*). Procedure (*a*) follows:—

(A) FOR STATIC BALANCE.

1. If the object to be balanced is not readily separable from its shaft, it will be necessary to balance the entire unit. If, however, it is a clutch flange or fly wheel it is desirable that it be mounted on an accurately fitted, perfectly balanced arbor.

2. Mount the object on the rolls of the balancing machine so that it is free to rotate and has positive thrust.

3. An accurately balanced clamp, of a size suitable for the work in hand, is then fitted to the shaft, usually through the intermediary of an accurately centered bushing. Any eccentricity in this particular would be fatal to accurate results. This clamp carries, in some types one, and in other types two, known weights. These are adjustable by movement away from a neutral position. When the weights are in the neutral position, the clamp is in perfect balance. Place the weights in the neutral position.

4. Lock the vibrating support or dynamic pendulum and free the static pendulum by proper manipulation of the screws provided to accomplish this.

5. By means of a motor, rotate the object to be balanced. When the driving motor is started, increase the speed of rotation slowly until the static pendulum oscillates smoothly or until the speed is known to be well above that required to cause any unbalance present to be in synchronism with its supports. When the synchronous speed is reached, the vibrations of the frame of the machine will be a maximum and will also become regular and uniform in amount as shown by the micrometer dial indicator. If the indicator does not show any vibration, it may be that the synchronous speed has not been reached, rather than that the object is in static balance. It should be borne in mind that the synchronous speed as obtained in the balancing machine is not necessarily that of the mass under operating conditions. It is usually much lower and is a function of the period of the pendulum acting as a support. This, in turn, depends upon the tension of the springs under the locking arms. In this connection, it is important to remember that, regardless of the synchronous speed so found, if the mass is balanced at that speed it will be in balance at any other speed. It is important, therefore, to establish whether the mass has not yet reached its synchronous speed or whether it is in static balance. To ascertain this, move the weight a short distance from its neutral position. This introduces a small amount of unbalance. If no vibration results, it is established that the synchronous speed has not been reached, *i.e.*, the free period of the frame does not correspond to the period of the object under test. It is then necessary to change the period of the support by varying the tension of the springs until synchronism results.

6. Assuming unbalance has been found to exist, it is first necessary to establish the plane of unbalance. To accomplish this, introduce, by means of the clamp weight, a positive unbalance at any arbitrarily chosen plane. If this known unbalance is in the plane of the inherent unbalance of the mass, vibration will be of maximum amplitude. If it is in the opposite plane, the amplitude will be a minimum. Intermediate planes will give amplitudes of intermediate values. A recommended procedure for determining the plane of unbalance is as follows:—

(*a*) Put the clamp weight in a position of known unbalance, by moving it away from the neutral position an arbitrarily chosen

distance. Do not change this distance from the axis of rotation until the plane of unbalance is definitely established.

(*b*) Secure clamp in any arbitrary plane. Regard this as the zero position.

(*c*) Rotate the object and record the amplitude of vibration at the synchronous speed.

(*d*) Repeat this for clamp positions or planes 90, 180 and 270 degrees from the initial position.

(*e*) Plot the readings obtained on a polar diagram. See Figure 8.

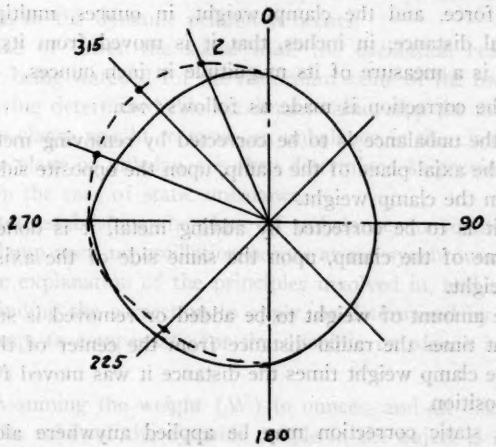


Fig. 8.

This shows the amplitude to be a maximum at 270 and a minimum at 90, curve 1, and that the plane of unbalance lies in the vicinity of the 270 degree plane.

(*f*) Set the clamp on the 315 and 225 degree planes. Plot the amplitudes obtained. Curve 2. Since the amplitude at 315 is greater and that at 225 is less than that at 270, it appears that the plane of maximum vibration lies near 315.

(*g*) Continue readings with new clamp settings until sufficient points are plotted to draw an accurate curve. The point of maximum deflection can be ascertained closely from this. Set the clamp on the predicted plane and check its correctness by rotating the mass.

(*h*) After finding the plane of unbalance, the amount of unbalance is obtained by moving the clamp weight toward or away from the center. The change in indicator readings denotes whether or not the change is in the proper direction. When the indicator is steady at the synchronous speed, the unbalance is equilibrated. The center line of the clamp indicates the axial plane of the disturbing force, and the clamp weight, in ounces, multiplied by the radial distance, in inches, that it is moved from its neutral position, is a measure of its magnitude in inch ounces.

(*i*) The correction is made as follows; *viz.*

1. If the unbalance is to be corrected by removing metal, it is done in the axial plane of the clamp, upon the opposite side of the axis from the clamp weight.

2. If it is to be corrected by adding metal, it is done in the axial plane of the clamp, upon the same side of the axis as the clamp weight.

3. The amount of weight to be added or removed is such that its weight times the radial distance from the center of the shaft equals the clamp weight times the distance it was moved from the neutral position.

4. The static correction may be applied anywhere along the length of the body. It is, however, preferably made in, or as near as possible to, the plane through the center of gravity normal to the axis of the body since at this point there is likely to be less disturbance of the dynamic balance.

5. After the correction has been applied, return the object to the machine for the dynamic balancing test.

In this method of balancing it is to be noted that the dynamic balance is based upon the static balance, and is good or bad depending upon whether the static balancing is well or poorly done.

After the static unbalance is corrected, proceed as follows to obtain:

(B) THE DYNAMIC UNBALANCE.

1. Lock the static pendulum and free the dynamic pendulum.

2. Secure the balancing clamp to the shaft with the weights in their neutral position. Its position on the shaft is immaterial, although it is usually placed near the vibrating support. In cases where a large dynamic unbalance is found, the clamp should be placed as far as possible from the stationary support, since its value for introducing a dynamic couple depends upon the distance of the weight from its neutral position, and the distance of the clamp from the center of the stationary support.

3. Rotate the mass, increasing its speed slowly, until a regular oscillation of the dynamic pendulum occurs.

4. Raise or lower the speed until the oscillation ceases, the indicator being watched for the maximum value of the oscillation.

5. Having determined this maximum value, stop the motor and move the clamp weight to cause an unbalance in the clamp.

6. The plane of unbalance is then determined as previously described in the case of static unbalance.

7. After establishing the plane of unbalance, move the weight on the clamp until no oscillation occurs at the synchronous speed.

8. The explanation of the principles involved in, and the manner of finding the correction to apply, as determined by the balance clamp, to compensate for the dynamic unbalance is as follows, *viz;*

(*a*) Assuming the weight (W) in ounces, and all distances in inches, the value of the dynamic couple in the object is given by the product of the weight, times the distance "W" is moved from its neutral position in the clamp, times the distance from the *center* of the clamp to the *center* of the stationary support. See Figure 9.

(*b*) The rotating body may be assumed to be made up of an indefinite number of elements each lying perpendicular to the axis, and having its own center of gravity. If all centers of gravity lie on the axis of rotation, no dynamic or static unbalance exists. Since this mass has been balanced statically, the center of gravity, C.G., of the mass must lie on the axis of rotation. As dynamic unbalance has been found to exist, it is evident that the centers

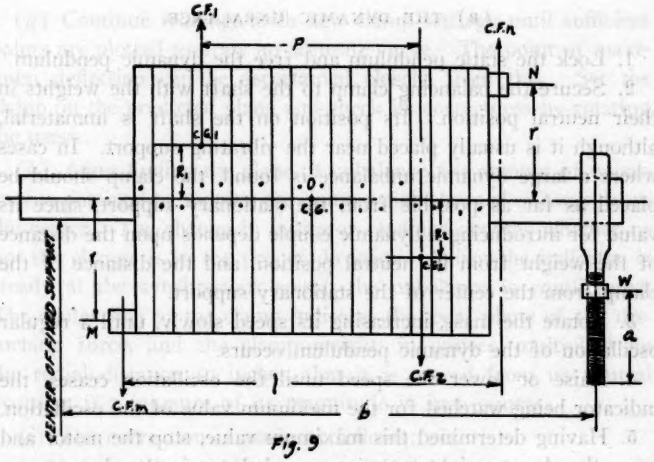


Fig. 9

of gravity of the sections to the right and left of "O" do not lie on this axis. They must, however, balance each other and may be represented as CG_1 and CG_2. When the mass is rotated CG_1 will produce a centrifugal force, $CF_1 = \frac{W_1 w^2 R_1}{g}$. The centrifugal force produced by CG_2 is $CF_2 = \frac{W_2 w^2 R_2}{g}$.

Since the body is in static balance, $W_1R_1 = W_2R_2$. Therefore, $CF_1 = CF_2$. Hence there are two equal and opposite centrifugal forces CF_1 and CF_2 acting at a distance p to produce a rocking moment $CF_1 \times p$.

(*c*) This centrifugal couple may be neutralized by adding or removing material at two points in the plane of unbalance such that the centrifugal couple produced thereby shall balance the other. It is not necessary that this material be added or removed at a distance equal to "p." It may be added or removed at any convenient points in the plane of unbalance. If it be added at points such as "M" and "N," it is necessary only that the moment due to the centrifugal couple of the weight at "M" be equal and opposite to the moment $CF_1 \times p$.

(*d*) If a weight M be put at a radius " r," which may be chosen arbitrarily, then $CF_m = \frac{Mw^2r}{g}$ where CF_m is the centrifugal force at " M."

Then $CF_1 \times p = CF_m \times 1$

or $W_1w^2R_1p = Mw^2r1$. w is common to both

Hence $W_1R_1p = Mrl$.

(*e*) That is, the product of *one* of the added weights, its radius or distance from the center of rotation, and the distance between the centers of gravity of the added weights must equal the corresponding product for the dynamic unbalance in the object. It follows, therefore, that if the product of W_1R_1p can be found, the weight to be added may easily be computed since " r" and "1" may be arbitrarily fixed. This product, W_1R_1p, in " inch square ounces," is found by the dynamic balancing machine as the product of the weight in ounces of the movable weight " W," the radial distance in inches this weight has been moved on the clamp from its neutral position, and the axial distance in inches of the center of the clamp from the center of the stationary support.

9. It is essential that M and N be of equal weight in order that a couple will be formed, and also that they are exactly 180 degrees apart so that the static balance, previously obtained, will not be destroyed.

Due to the number of operations and the length of time involved in ascertaining and correcting static and dynamic unbalance separately the following more recent procedure is most generally employed. It consists in eliminating, in one operation, the effects of dynamic and static unbalance when the mass is rotated. Since static unbalance is of no consequence when the mass is stationary, it is necessary only to correct the result of its centrifugal force during rotation of the mass. As before shown, dynamic unbalance occurs only when the mass is rotated consequently, if the resultant of these two conditions, as manifested when the mass is in motion, is corrected, the desired end is attained, *i.e.*, rotation without vibration.

This method of balancing depends upon two theorems relative to forces and couples acting on a rigid body—

(*a*) Any number of forces acting on a rigid body is equivalent to a force and a couple.

Proof:—(See Figure 10)

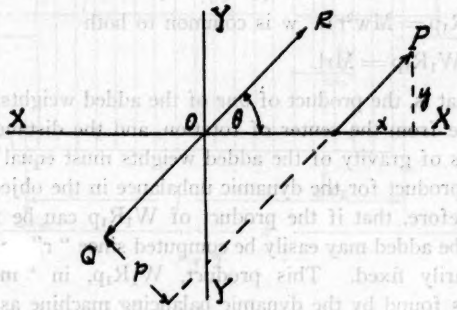


Fig. 10

If:—

P = Resultant of all forces acting on the body.

x = Resultant of all forces acting along the X axis.

y = Resultant of all forces acting along the Y axis.

$P = \sqrt{x^2 + y^2}$ and $\tan \Theta = y/x$.

Θ = angle P makes with the X axis.

Pi = original forces.

pi = perpendicular distance from origin to the point of application of the forces.

These forces acting through moment arms form couples.

Their summation Σ Pi pi = Σ Ci. The resultant of Ci = C.

The equal and opposite forces, R and Q, may be introduced without effect on the body, if they pass through O and are equal and a parallel to P. By their introduction, a single force R and a couple Pp exist. Their effect on the rigid body is the same as that of the original force P.

(*b*) The resultant of a couple and a force parallel to the plane of the couple, is a force equal in magnitude and parallel to the original force, at a distance from it equal to the moment of the couple divided by the magnitude of the force. (See Figure 11).

Fig. 11.

A couple may be transferred to any part of a rigid body so long as its magnitude remains constant and its direction remains the same.

If P = a force parallel to the plane of Qx, a couple, the resultant is a force,

$P_1 = P$, acting at a point A, so that

$P_1 y = Qx$ = moment of couple

$$\text{or } y = \frac{Qx}{P}$$

The application of these laws of statics follows.

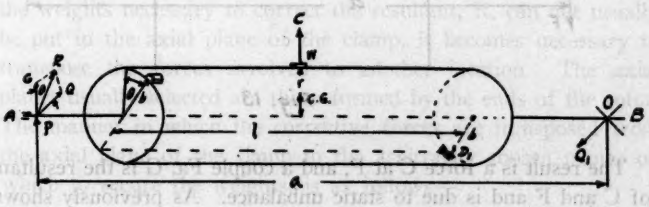


Fig. 12

Figure 12 represents a rotor in static and dynamic unbalance.

W = a weight at a distance r from the axis AB. It is in a plane perpendicular to the axis of rotation passing through the center of gravity of the rotor. This represents static unbalance which, by virtue of rotation, produces a centrifugal force, C., acting in the direction shown.

D and D_1 are equal weights, located a distance b, from the axis of rotation, and 1 apart, lying in a plane passing through AB. They are 180 degrees apart. Due to the rotation of the armature D and D_1, produce a couple $\frac{Dbw^2l}{g}$. This represents the dynamic unbalance of the rotor.

If now "O" is taken as the fixed bearing, it is possible to find a force Q acting at P in the plane of DD_1, equivalent to the couple

$$\frac{Dbw^2l}{g}.$$

The static unbalance Wr becomes a centrifugal force, $C=\frac{Ww^2r}{g}$, upon rotation of the mass. This force may be transferred to P, parallel to itself, but in so doing a couple Fa must be introduced. (See Figure 13.)

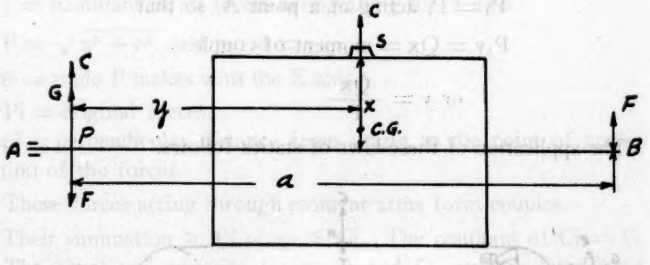


Fig. 13

The result is a force C at P, and a couple Fa. G is the resultant of C and F and is due to static unbalance. As previously shown in Figure 12, Q is the force due to dynamic unbalance. These

forces act at P and may be combined to give as a resultant a single force R, acting the direction and of the magnitude shown graphically in Figure 14.

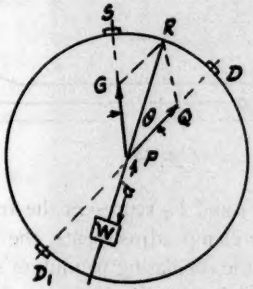


Fig 14

The plane and magnitude of R is ascertained by means of the balance clamp as previously explained. It should be noted that the balance clamp weight lies directly opposite the disturbing force R. When the weight is so located that no vibration occurs, all the unbalanced forces existing with the "B" end fixed are equilibrated. This condition can be made permanent by the substitution of proper weights when the clamp is removed.

Before the balancing is completed, however, it is necessary to "fix" the A end and allow the B end to oscillate. The same procedure is followed to compensate for unbalanced forces. Since the weights necessary to correct the resultant, R, can not usually be put in the axial plane of the clamp, it becomes necessary to transpose the forces involved to another location. The axial planes usually selected are those formed by the ends of the rotor. The manner in which the corrective forces are transposed from the axial plane of the clamp to the arbitrarily chosen planes on which to secure the weights, is as follows—

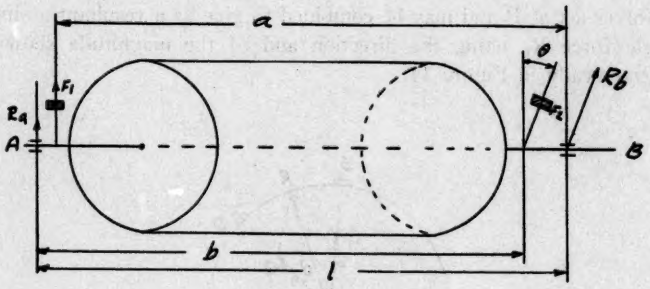


Fig. 15.

In Figure 15 let F_1 and F_2 represent the forces, in inch ounces, corresponding to the clamp adjustments, the former when A and the latter when B is the oscillating pendulum support. The forces at the supports A and B will then be

$$Ra = \frac{F_1 a}{l} \text{ and } Rb = \frac{F_2 b}{l}.$$

Their directions will be parallel to F_1 and F_2 respectively.

According to the foregoing, a force Ra, introduced at A, and a force Rb, introduced at B, will equilibrate the forces due to unbalance. These are shown in Figure 16 where I and II are the flanges at which corrections must be made.

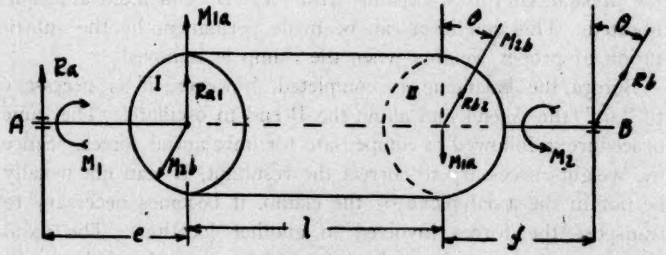


Fig. 16

By statics the force Ra can be transported, parallel to itself, to flange I, provided the moment M_1, is introduced. $M_1 = Ra \times e$. The force so transported is indicated by Ra_1. The moment M_1

can be replaced by a couple formed by two forces M_1a acting in the plane of M_1, and therefore parallel to Ra, at the flanges I and II as shown. The magnitude of $M_1a = \frac{M_1}{l} = \frac{Ra \times e}{l}$.

In a similar manner, the force Rb is transported to flange II and is indicated as Rb_2. As in the previous case the moment, M_2, must be introduced. It is equal to $Rb \times f$. This moment is, in turn, replaced by two forces, M_2b, acting in the plane of the moment M_2. These forces are parallel to Rb and form a couple

$$M_2b = \frac{M_2}{l} = \frac{Rb_2 \times f}{l}$$

By the foregoing operations, the forces at the bearings have been replaced, without disturbing the equilibrium, by 3 forces each at the flanges I and II. These latter, therefore, must also equilibrate the forces of unbalance.

The last step in the solution consists in determining the above forces analytically, or, as is shown in Figure 17, graphically.

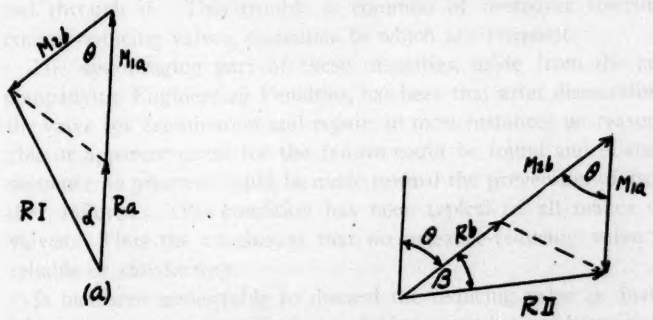


Fig. 17.

In Figure 17 (*a*) and (*b*), RI and RII are the resultants and their position on the body is determined by the angles, α and β. The angle Θ is the angle between the clamp positions, Figure 16. Hence, if at the axial planes I and II, either by adding or removing weight, the forces RI and RII are introduced in the radial

planes defined by the angles α and β, all unbalance, static as well as dynamic, will be eliminated.

Should it be desired, for any purpose, to separate the static and dynamic corrections, Ra and Rb represent the static forces while the dotted lines in Figure 17 (*a*) and (*b*) represent the dynamic forces.

This method has the advantage of not requiring the determination of the center of gravity of the body, and gives the necessary corrections for unbalance as a whole without distinguishing static or dynamic unbalance.

It is hoped that this article will assist in a better general understanding of the subject of balancing, that it will result in the accomplishment of balancing where necessary and in the saving of funds by accomplishing the work more expeditiously as well as by avoiding the undertaking of unnecessary work.

PRESSURE REDUCING VALVES.

By Lieut. H. C. Rule, Jr., Member.

Considerable trouble has been experienced in the Service afloat with the operation of pressure-reducing valves for air, steam and water. This is particularly true of the steam pressure-reducing valve which has been generally conceded in the Service as unreliable in its present state of development.

This lack of confidence in the reducing valve is the natural result of the many failures of this type valve with no apparent cause for such failure. In many cases, the valve had been thoroughly overhauled shortly prior to the failure and great care had been taken in its reassembly. Because of repeated instances of this kind, little faith is placed in the valve by the operating personnel and trouble is generally expected from the machinery units fed through it. This trouble is common of destroyer steering engine reducing valves, casualties to which are frequent.

The discouraging part of these casualties, aside from the accompanying Engineering Penalties, has been that after dismantling the valve for examination and repair, in most instances no reasonable or apparent cause for the failure could be found and in consequence no progress could be made toward the prevention of further failures. This condition has been typical of all makes of valves. Thus the conclusion that no pressure-reducing valve is reliable or satisfactory.

It has been undesirable to discard the reducing valve in favor of operating certain auxiliaries on high-pressure steam because of the increase in weight of the units and their comparative inefficient operation at the usual low cruising load. So an exhaustive study of the situation has recently been made in cooperation with designers and manufacturers of these valves.

As this study progressed, it developed that the past method of specifying the size of reducing valves was in error. In short, the past method was as follows:—The required size of steam line for

supplying the auxiliaries operated at reduced pressure was computed for full load conditions. This line would accommodate the maximum steam supply at a predetermined velocity demanded for supplying the auxiliaries under the maximum load requirements. Since it was required that a reducing valve be placed in the line, a valve of the same size as the line was specified. The high pressure steam line leading to the reducing valve was also specified the same size as the reducing valve in order that the flanges of both sections of line and of the reducing valve would match. See Figure 1.

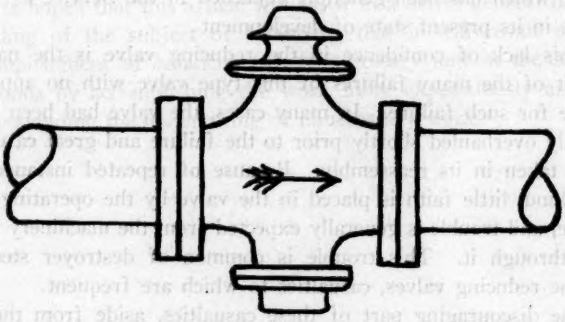


FIGURE 1.

Thus it may be seen that the high-pressure line supplying steam to the reducing valve was capable of supplying much more steam than could possibly be consumed by the auxiliaries as the same size line was capable of sufficient supply at the reduced pressure. It may further be seen, that the reducing valve, in this case, was being operated at the extreme end of its range as such a large valve passing such a comparatively small amount of steam would have to be set close to the shut-off position. This was a very unfair condition imposed upon the valve as any reducing valve will operate to its best advantage in the middle of its pressure range.

Thus there were two evident faults in the past requirements: (1) that the steam line leading to the reducing valve was considerably larger than was necessary, thus adding excess weight, and

(2) that the valve was not being operated in the proper range for which it was designed. The latter fault caused practically all the failures that could be attributed to design. The common chattering effect accompanied by erratic fluctuations in steam pressure on the discharge or low pressure side and occasionally a sticky valve are the results of the installation of too large a valve.

After having determined the above, the proper method of specifying the correct size of reducing valve was developed. It has been proven by experiment that this new method will produce results that have been previously hoped for but never, or seldom ever, attained. The method is as follows:—Determine the area of the line that will pass the required pounds of steam per hour at a velocity of 6000 feet per minute to supply the auxiliaries at the reduced pressure. This will be the area of the reducing valve discharge line which is the line on the low-pressure side of the reducing valve and feeding the auxiliaries. Obtain from the steam tables the specific volume of the steam at this exhaust-side line pressure. Then obtain the specific volume of the steam at the designed main steam line pressure. The ratio of the area of the line on the high pressure side of the reducing valve to the area of the line on the low pressure side of the reducing valve will be directly proportional to the ratio of the specific volumes of the steam in these lines. This will give the size of the line required to supply the proper amount of steam through the high pressure line to the reducing valve. Then the nominal size of the reducing valve will be the same as the nominal size of the high pressure line leading to the valve.

For example let us consider the size reducing valve required and the size of high pressure steam line required to supply auxiliaries operating at a steam pressure of 30 pounds per square inch gauge and requiring 25,000 pounds of steam per hour. The velocity of the steam must not exceed 6000 feet per minute and the main steam line pressure is 300 pounds per square inch gauge.

Spec. Vol. steam at 30 pounds per square inch is 9.39 cubic feet per pound.

Spec. Vol. steam at 300 pounds per square inch is 1.479 cubic feet per pound.

Then 25,000 pounds per hour × 9.39 cubic feet per hour is 234,750 cubic feet per hour required.

And 6,000 feet per minute × 60 is 360,000 feet per hour velocity.

Then 234,750/360,000 gives .652 square feet area of exhaust line or 11 inches diameter.

And .652 : × :: 9.39 : 1.479 or × equals .1027 square feet area of the supply line or 4.34 inches diameter.

Therefore, the supply line to the reducing valve and the reducing valve itself will be specified at 4½ inches which is the nearest size valve above 4.34 inches.

From the above deduction, it may be seen that a 4½ inch inlet to the reducing valve will be required, while the outlet must be 11 inches. This hook-up is accomplished by using a 4½ inch reducing valve with the flanges on the inlet and outlet ends the same size (this size being the required flange size for a 4½ inch reducing valve) and connecting the valve to the low-pressure line by means of a taper expander. This expander is specified in all cases as being 2 feet in length.

The results of the above arrangement were very encouraging but a further modification was suggested. Attention was called to the fact that the velocity of the steam in the discharge side of the valve is very high and that the flow of the steam through this section is very turbulent. The pressure at the discharge throat of the valve is therefore undoubtedly different to a varying degree from that in the expanded discharge line after the steam has been expanded and settled down to a condition of steady flow. Because of the fact that the fluctuations in pressure are not as noticeable in the expanded line as they are in the throat of the valve, the valve will operate at a much steadier and more constant pressure if its governing diaphragm be actuated from the expanded line instead of from the discharge throat of the valve as is now the custom. The exact pressure in the low pressure line can thus be controlled instead of assuming that it is the same as in the throat of the valve. So the method of remote control was introduced with further gratifying results. See Figure 2.

However, all of the failures of the reducing valve in Service cannot be blamed on design and improper installation because the

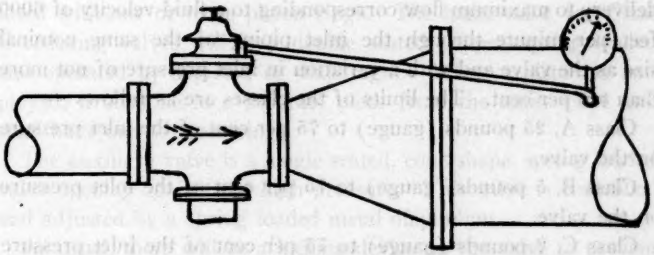


FIGURE 2.

question of material plays a very important part in the operation. The material side of the story was given very thorough study and further experiments were conducted along these lines. It was found that, with this new type of valve employing the proper materials in construction, a greater pressure range could be obtained and thus one valve could accomplish the pressure reduction which previously would have required two valves. Also valve failures due to sticking could be practically eliminated. So it may be seen that quite satisfactory results are expected of the new type valve when properly installed and of the correct size and class.

Some of the most important features of the new design are described hereinafter so that a definite idea of the product may be visualized. The valves are required to be of the auxiliary operated type. That is, with a secondary or pilot valve which is controlled from the discharge line and operates the main valve. They are divided into three classes according to the maximum initial pressure and temperature for which they are suitable. Class A calls for a valve satisfactory for use with steam pressure of 450 pounds per square inch (gauge) and a temperature up to 710 degrees Fahrenheit. Class B is required to be suitable for 300 pounds steam pressure and a maximum of 425 degrees temperature, while the Class C is for 100 pounds pressure and a maximum of 425 degrees Fahrenheit.

However, each type is required to be capable of satisfactory operation in pressure reduction when set to deliver any pressure within the range for its class and between limits of flow from no

delivery to maximum flow corresponding to a fluid velocity of 6000 feet per minute through the inlet piping of the same nominal size as the valve and with a variation in inlet pressure of not more than ten per cent. The limits of the classes are as follows:—

Class A, 25 pounds (gauge) to 75 per cent of the inlet pressure on the valve.

Class B, 5 pounds (gauge) to 75 per cent of the inlet pressure on the valve.

Class C, 2 pounds (gauge) to 75 per cent of the inlet pressure on the valve.

In all classes, this reduction will be accomplished by a single adjustable spring and diaphragm.

The inlet and outlet connections are specified the same size whether flanged or screwed and the expanded area in the low pressure line shall be provided for separately by the user of the valve to meet the actual need in installation.

The pipe or tube by means of which the secondary or pilot valve is operated from the discharge line is required to be of not less than ¼ inch iron pipe size and not less than three feet in length from the outlet flange of the valve to the discharge line connection. This is to insure that the pressure that actuates the valve proper is really the pressure existing in the low pressure line. It is further required that this pipe or tube be so installed that any condensation that takes place in it will flow from the valve into the discharge line thus keeping the tube constantly clear of water and insuring proper actuation of the valve diaphragm. There will also be a gauge connection at the point where this tube enters the low pressure line so that the exact pressure in that line may be determined by inspection and the exact setting of the valve may be assured.

There is a requirement that it shall be possible by manual operation to render the automatic reducing feature of the valve inoperative and thus allow a full flow through the valve. By this feature the machinery units fed through the valve may still receive a supply of steam should the valve stick closed.

The main valve of Class A and B valves is to be single seated poppet, spring loaded and operated by a piston. The piston is required to be in a separate piece from the main valve and to be

fitted with one or more piston rings. The main valve for Class C reducing valves will be a double seated poppet, spring loaded and operated by a piston or diaphragm. When a piston is employed, it shall be in a separate piece from the main valve and shall have one or more rings.

The auxiliary valve is a single seated, cone shape, spring loaded valve opening against the high pressure. This valve is controlled and adjusted by a spring loaded metal diaphragm.

As usual, there will be a locking device on the valve to prevent unauthorized change in adjustment of the spring which controls the auxiliary valve. Also the auxiliary valve chamber, high pressure side and diaphragm chamber will be self-draining.

Pipe flanges when required, will be finished on the face and edges and shall conform in diameter and thickness to the table given later in this article, unless special sizes are specifically required. These flanges shall not be drilled and for Class A valves shall be finished to provide for a metal to metal joint.

The design and construction of the entire reducing valve is required to be such as to afford easy access for repairs and overhaul without removal of the valve from the line.

The details in construction of the valve are in many cases radically different from the past method. There has been considerable difference of opinion as to where and when valves shall be flanged. This detail is specifically covered by the requirements of the new design. All Class A, B, and C valves above 1½ inches (nominal size) will be flanged throughout and Class B and C valves of 1½ inches or smaller will be flanged throughout but the top and bottom body bonnet for the main valve chamber may be screwed. The Class C valve of 1½ inches or smaller may have screwed connections throughout.

Through-bolts or studs will be used for the top and bottom bonnet flanges when such are flanged. Square ended stud bolts are required to be used on Class A valves.

In all classes of valves, renewable main and auxiliary valve seats and cylinder liners will be supplied.

The material for each class of valve will be given in the following table:

CLASS A VALVES.

Valve body 2 inches and smaller—Steel (Composition CN-r or CN-c bonnets may be used).

Valve body above 2 inches—Steel throughout.

Springs—Corrosion resisting steel or steel protected against corrosion by an approved method.

Diaphragms—Composition CN-r (46M7) or Composition P-r (46B14).

Piston rings—Cast iron, unless otherwise approved.

Cylinder liner—Composition CN-c, CN-r, or an approved corrosion resisting steel.

Bolts, studs, etc.—Class B or C steel (reB11).

Main and auxiliary valves shall employ an approved corrosion resisting steel in either disc or seat with Composition CN-c or CN-r used against the corrosion resisting steel in the other element.

Other parts—As recommended by the manufacturer and approved.

CLASS B VALVES.

Valve body 3 inches and over—Composition G (46M6).

Valve body under 3 inches—Composition G or M (46B8).

Springs—Corrosion resisting steel or steel protected against corrosion by an approved method.

Diaphragms—Composition CN-r (46M7) or Composition P-r (46B14).

Piston rings—Cast iron, unless otherwise approved.

Cylinder liner—Composition CN-c, Composition G or M, or an approved corrosion resisting steel.

Bolts, studs, etc.—Class B or C steel (43B11).

Main and auxiliary valves shall employ an approved corrosion resisting steel in either disc or seat with Composition CN-c, CN-r, or Composition G used against the corrosion resisting steel in the other element.

Other parts—As recommended by the manufacturer and approved.

CLASS C VALVES.

Valve body 3 inches and over—Composition G (46M6).

Valve body under 3 inches—Composition G or M (46B8).

Springs—Corrosion resisting steel or steel protected against corrosion by an approved method.

Diaphragms—Composition CN-r (46M7) or Composition P-r (46B14).

Piston rings—Cast iron, unless otherwise approved.

Cylinder liner—Composition CN-c, Composition G or M, or an approved corrosion resisting steel.

Bolts, studs, etc.—Class B or C steel (43B11).

Main and auxiliary valves shall employ Composition M or any of the valve materials specified under Classes A and B. Valve seats and discs may be of the same or dissimilar materials.

Other parts—As recommended by the manufacturer and approved.

As there has been considerable misunderstanding regarding the terms "nominal size," "iron pipe size," "commercial iron pipe size," and "size of tubing," and as there is a difference between the sizes of commercial flanges and the sizes of Navy Standard flanges for the same size pipe, the following table of flange dimensions with outside diameters of tubing for inlet and outlet connections of reducing valves is inserted as a guide to those interested.

The tests and inspections to which the new design of valve will be submitted are in some detail quite different from the tests heretofore required of the reducing valve. No reducing valve will be approved for Naval use which has not been submitted for test and inspection at a Navy laboratory and in addition a six months' service test on board a Naval vessel. At the place of manufacture, the valves will be inspected to determine compliance with the Navy Department specification relative to material, workmanship, and dimensions; and each valve body will be tested hydrostatically to 1½ times the maximum initial pressure designated by its Class. One completely assembled valve of each size and type from each group of ten or fraction thereof will be selected and subjected to a test to demonstrate fully its satisfactory operation. The tests will be as follows:—All valves of each type shall be subjected to

Nominal valve size	Outside diameter of tubing	Minimum Valve Capacity lbs. per hr. at 300 lbs. Pressure (Gauge)	Class A		Class B		Class C	
			Diameter	Thickness	Diameter	Thickness	Diameter	Thickness
Inches	*Inches*	*Pounds per hour*	*Inches*	*Inches*	*Inches*	*Inches*	*Inches*	*Inches*
¼	0.540	150	3⅜	9/16	3⅜	⅛	3¼	⅛
⅜	.840	550	4	9/16	4	⅛	3 9/16	⅛
½	1.050	950	4 5/16	⅛	4 5/16	1/16	3 13/16	1/16
1	1.315	1,600	5 1/16	¼	5 1/16	1/16	4¼	1/16
1¼	1.660	2,700	5⅝	⅛	5⅝	1/16	4¾	1/16
1½	1.900	3,700	5 15/16	1/16	5 15/16	¼	5 1/16	1/16
2	2.375	6,000	6½	¼	6½	¼	5 9/16	1/16
2½	2.875	9,000	7 9/16	1/16	7 9/16	1/16	6⅛	⅛
3	3.500	13,000	8½	1/16	8½	1/16	6⅝	⅛
3½	4.000	17,000	8 11/16	⅜	8 11/16	⅜	7 3/16	⅛
4	4.500	22,000	9½	1/16	9½	⅜	7 11/16	⅛
4½	5.000	27,000	9 13/16	1	9 13/16	1/16	8 3/16	⅛
5	5.563	35,000	10⅜	1	10⅜	1/16	9 1/16	1/16
5½	6.125	42,000	11⅜	1 1/16	11⅜	1	9 9/16	1/16
6	6.625	47,000	11 13/16	1⅛	11 13/16	1	10⅛	1/16
6½	7.125	56,000	12 9/16	1 3/16	12 9/16	1 1/16	10⅝	1/16
7	7.625	65,000	13⅛	1 3/16	13⅛	1 1/16	11 5/16	1/16
7½	8.125	72,000	14 3/16	1¼	14 3/16	1⅛	11¾	⅛
8	8.625	81,000	14¾	1 5/16	14¾	1⅛	12⅜	⅛
8½	9.125	90,000	15 5/16	1⅜	15 5/16	1 3/16	12 15/16	⅛
9	9.625	100,000	15⅞	1⅜	15⅞	1 3/16	13 7/16	1/16
9½	10.250	115,000	16 7/16	1 7/16	16 7/16	1¼	14¼	1/16
10	10.750	130,000	17	1¼	17	1¼	15	1/16

the Dead End Condition test. In this condition the delivered pressure shall not vary from the reduced pressure setting by more than plus or minus ten pounds for a period of one hour. The dead end shall be not less than ten feet or fifty diameters from the valve. Under all other than dead end conditions, and for a reduced pressure setting of 25 pounds and above, the delivered pressure shall hold between limits of plus or minus five per cent of the reduced pressure setting while the capacity demand is varied. There will be no hunting permitted under steady flow conditions. For all sizes of valves up to and including 2½ inches, the satisfactory operation of the valve shall be demonstrated at the full designed pressure and capacity. For all valve sizes from 2½ inches to 4 inches a reduced capacity to within the steam supply of the manufacturer will be allowed. The reduced capacity determination shall be agreed upon by the inspector and the manufacturer and referred to the Bureau for approval.

From the foregoing design requirements, it may be seen that when ordering a valve, misunderstanding between the party drawing the requisition and the contractor may be averted if the requisition states the pressure class of the valve, the nominal size of the valve, the expected inlet pressure, the flange dimensions of the inlet piping, and the required capacity of the valve at 300 pounds pressure (gauge). It must be borne in mind that Class B and C valves are not suitable for use with superheated steam of a temperature in excess of 425 degrees Fahrenheit and that valves supplied for use with superheated steam of a temperature in excess of 425 degrees Fahrenheit should be designated as Class A.

GEORGE FISCHER CAST STEEL IN SUBMARINE DIESELS.

By LIEUTENANT JOHN O. HUSE, U. S. N., MEMBER.

PART I.

INTRODUCTION.

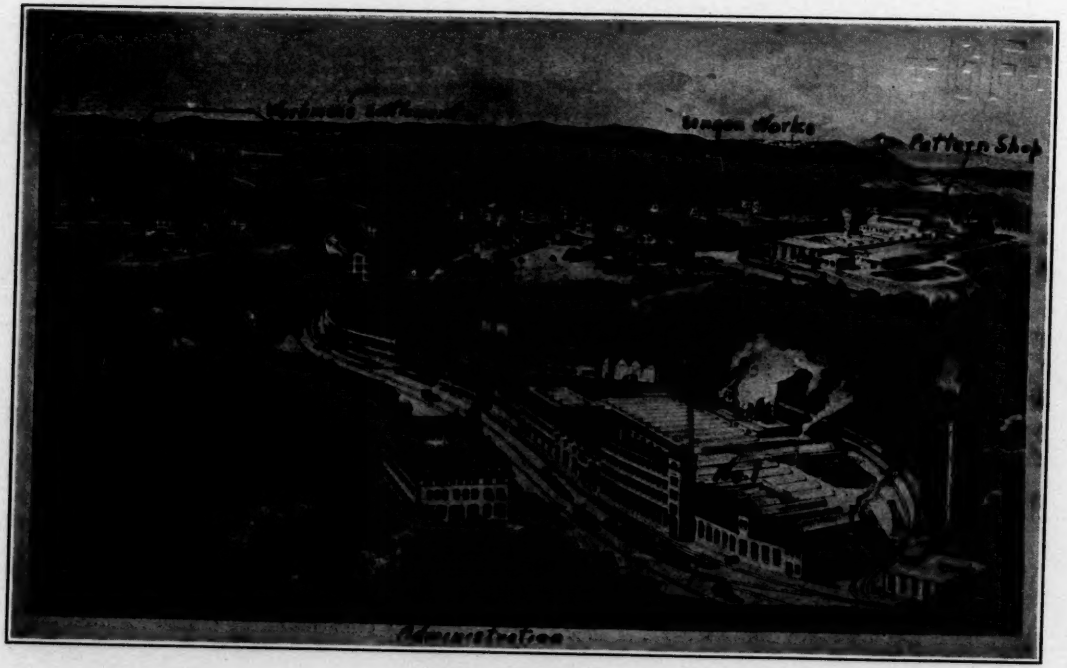
The cooperation and the experience of a first-class steel foundry are essential for the manufacture of Diesel engines. The design of high-speed types usually requires large, flawless steel castings with a thin section. The framing, as well as the cylinder jackets, are often cast steel. The design is, therefore, highly sensitive to poor foundry work, which would cause great expense and loss in dependability, as well as increase in engine weight. These high-class steel castings required for the manufacture of Diesel engines have been produced for a number of years at the steel foundries of the Société Anonyme des Aciéries ci-devant Georges Fischer, in Schaffhausen, Switzerland. This is a fact well known to engineers.

The United States Navy experienced the greatest difficulties in attempting to duplicate Fischer work. In any case, Fischer is a name of great interest to Diesel manufacturers.

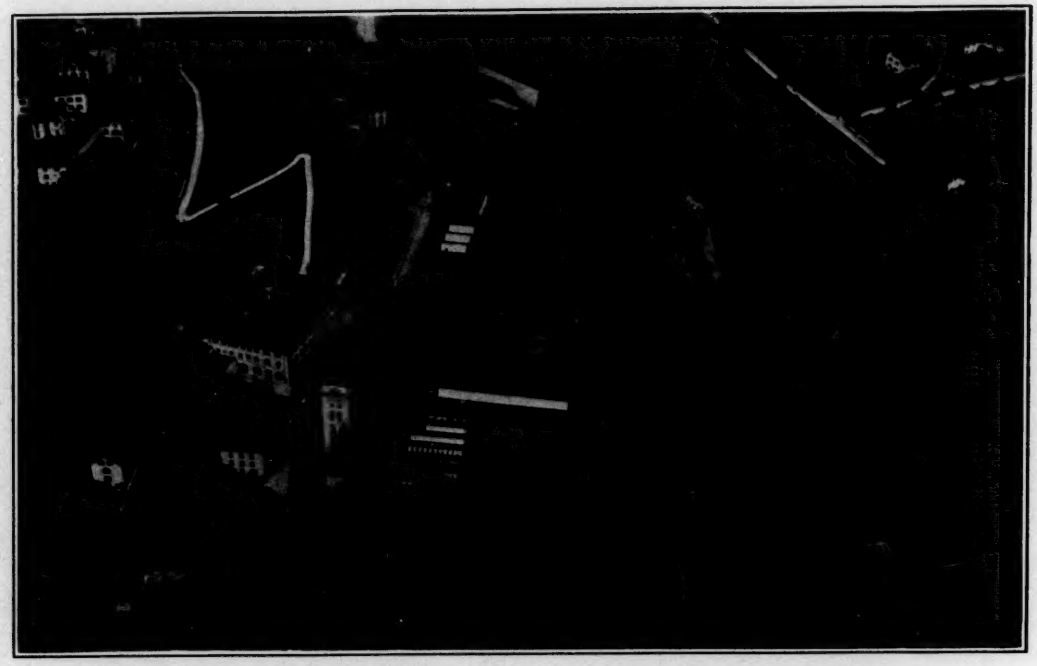
The reader may be unfamiliar with foundry practice. Therefore a general description, giving some of the reasons for steel casting difficulties, may be of value.

Study of the following equilibrium diagram illustrates clearly the troubles met in casting steel. The diagram is not accurate because the temperatures are those obtained on heating. Also the diagram represents the pure iron-carbon system. However, it is believed to be sufficiently accurate to illustrate the causes for the worries of the steel casting foundryman:

(1) The melt commences to freeze when its temperature is lowered to the liquidus line or ABC. In the case of low carbon



GEORGE FISCHER PLANT.



GEORGE FISCHER CAST STEEL—SHOP NO. 1 FOR LARGE STEEL CASTINGS OVER 250 LBS.

steel (XY), the liquidus temperature is several hundred degrees higher than the liquidus temperature of cast iron (line X^1 Y^1). This fact means that the metal must be poured very hot so that it will run into the remote parts of the mold without freezing. The prevention of these cold shuts is very difficult in large castings, where the metal has a long path to run in cold sand. It is sometimes impossible where thin sections are demanded.

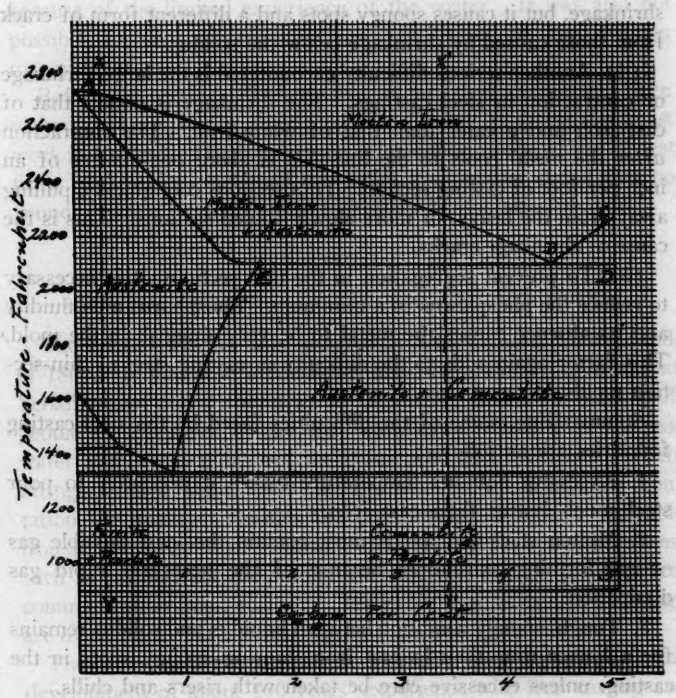


FIG. 1. EQUILIBRIUM DIAGRAM FOR IRON-CARBON ALLOY.

(2) The metal cools to the liquidus more quickly because of: (1) the larger temperature differences between sand and metal, and (2) the very thin sections demanded in Diesel castings for lightweight engines. Also, because of the small temperature drop

GEORGE FISCHER CAST STEEL—SHOP NO. 1 FOR LARGE STEEL CASTINGS

between lines AB and AE in the figure, the metal solidifies rapidly. Hence the gas contained in the melt in the spongy state during cooling from the liquidus, AB, to the solidus, AE, must be rapidly dissipated. This leads to blow holes and flaws unless great care is taken to give the mold sand porosity.

(3) In the semi-solid state between liquidus, AB, and solidus, AE, the metal is a tough spongy mass which pulls away from the colder and stronger sections. This is a form of contraction or shrinkage, but it causes spongy spots and a different form of crack from those caused by shrinkage below the solidus.

(4) Another serious difficulty encountered is the large shrinkage or contraction in steel castings. The shrinkage is double that of cast iron and greater than that of white iron. The contraction *after* the metal cools to the line AE is about one-quarter of an inch per foot of casting section. Thin sections cool quickly, pulling away from the heavy, or more slowly cooled sections. This is the cause of shrinkage cracks.

(5) To prevent segregation in the thin sections, it is necessary to reduce the phosphorus to a minimum. Phosphorus aids fluidity and its absence causes the metal to be very sluggish in the mold. This factor again adds to the difficulty in casting steel in thin-section molds of large sizes.

Summarizing, some of the difficulties faced by the steel casting foundries are as follows:

1. Excessive liquidus temperature makes it necessary to pour steel much hotter than cast iron.

2. Molten steel and molds must contain the least possible gas or gas-forming substances because of the necessary rapid gas dissipation.

3. Tough, spongy condition between liquidus and solidus remains for a longer time than in cast iron, causing spongy spots in the casting, unless excessive care be taken with risers and chills.

4. Contraction and shrinkage below the solidus is twice that of cast iron.

5. Very low phosphorus causes reduction in fluidity.

The importance of steel castings to the Diesel engine has been pointed out. The causes of trouble in casting steel as compared

with iron have been briefly discussed. The following pages are devoted to the ways and means employed by George Fischer to obviate these difficulties in order to produce sound, thin-section steel castings for the Diesel industry.

The writer's experience with foundries is limited. In describing the foundry practice at Fischer, therefore, everything observed has been noted, even though it may chance to be normal practice in every foundry. Undoubtedly, many points were overlooked because of the limited experience of the writer. It was also not possible to demand too much of the time of the Fischer management.

Before inspecting Fischer, five other steel foundries were visited. The writer is convinced that Fischer is doing the finest work in Europe. The foundry superintendent of one of the largest firms in America confirmed this belief and further stated that no foundry in the world is equal to Fischer in this class of work.

PART II.

GENERAL DESCRIPTION OF THE GEORGE FISCHER WORKS.

The firm of Fischer was founded at Schaffhausen, Switzerland, in 1802. It has now 5 foundries and one engineering works at Schaffhausen and 2 foundries in Germany; altogether over 8,000 people are employed by the firm. Malleable iron fittings for gas, water and steam pipes are the chief product of Fischer. Their fittings, made in about 9,000 patterns, have earned an enviable reputation for quality. Besides the pipe fittings, Fischer's malleable iron foundries produce malleable iron casting for many purposes, such as castings for the automobile industry, fittings for electrical communication lines, etc. . . . Malleable iron fittings and castings are of no particular interest within the scope of this article.

The foundry practice used in Fischer's steel foundries is the development of about 100 years' experience with iron casting and about 40 years' experience with steel casting.

The experience has been gained under ideal conditions. The laboratories and experimental projects have been plentifully supplied with funds. The quality of Fischer laborers is excellent. The laborers are largely natives of Schaffhausen and vicinity who have been in Fischer employ for many years. The stability of the

work load at Fischer gives her personnel a permanency not possible in more volatile industries.

These conditions aid Fischer in training and retraining the highly skilled labor necessary to her steel casting foundries.

STEEL CASTING FOUNDRIES.

The three shops of interest, all located at Schaffhausen, are as follows:

Shop No. 1, the original steel foundry, is devoted exclusively to the manufacture of large castings, over 250 pounds in weight.

Shop No. 4 is devoted to small castings, less than 250 pounds in weight.

Shop "Ebnat," in operation only two years, is turning out small steel castings in quantity lots with the most modern equipment and production methods obtainable.

PART III.

MELTING PRACTICE AND THE CHARGE.

All three shops employ Héroult electric furnaces for melting the charge. This method of melting is more expensive than the basic open hearth, or others, but the resulting steel is far cleaner and more uniform than that produced by any other method. The following is a list of various melting processes for steel castings from best to poorest:

1. Electric Furnace.
2. Crucible Steel.
3. Acid Open Hearth.
4. Basic Open Hearth.
5. Acid Bessemer.
6. Basic Thomas.

Electric furnace steel is more expensive, not only because of the high cost of current where water power is not available, but also because its charge consists of high-grade "foreign" scrap (selected steel purchased in the open market) and of "domestic" scrap (risers and turnings from the foundry itself). The pig iron and additions

to the charge are high quality. In other words, the furnace must be charged with the most expensive materials.

Electric furnace steel is best for steel casting work because:

1. The melt is finished and refined under a deoxidizing slag and the resulting metal is very pure and uniform.

2. Electric heat permits the high temperatures necessary to increase the fluidity for thin-section work.

3. It is possible commercially to reduce sulphur to .025 per cent and phosphorus to .020 per cent. These figures cannot be equalled commercially by any other process in high-capacity work.

4. The mechanical properties of Fischer Electric Steel Castings are excellent, as is shown by the adjoined figures and graphs.

MECHANICAL PROPERTIES OF +GF+ ELECTRIC STEEL CASTINGS.

LOWEST FIGURES GUARANTEED.

Tensile Strength		Yield Point		Elongation per cent		Impact	
kg/mm²	tons/sq. in.	kg/mm²	tons/sq. in.	Gauge Length 5 × Dia	Gauge Length 10 × Dia	kgm./cm²	foot pounds per sq. in.
		Electric Steel Castings annealed					
40–50	25–32	20–25	13–16	26–20	22–18	8	373
45–55	29–35	22–27	14–17	23–17	20–15	6	280
50–60	32–38	25–30	16–19	20–14	18–12	4	187
55–65	35–41	27–32	17–20	17–12	15–10	4	187
60–70	38–45	30–35	19–22	14– 9	12– 8	3	140
		Electric Steel Castings heat treated					
40–50	25–32	26–33	17–21	32–26	25–21	10	467
45–55	29–35	29–35	18–22	29–23	23–18	8	373
50–60	32–38	33–39	21–25	26–21	21–16	6	280
55–65	35–41	35–41	22–26	23–19	18–14	6	280
60–70	38–45	39–45	25–29	21–17	16–12	6	280

Tensile Strength and Elongation
of
+GF+ Electric Steel Castings
compared with
British, American and German Standard Specifications
(Gauge Length for Elongation: 2 in.)

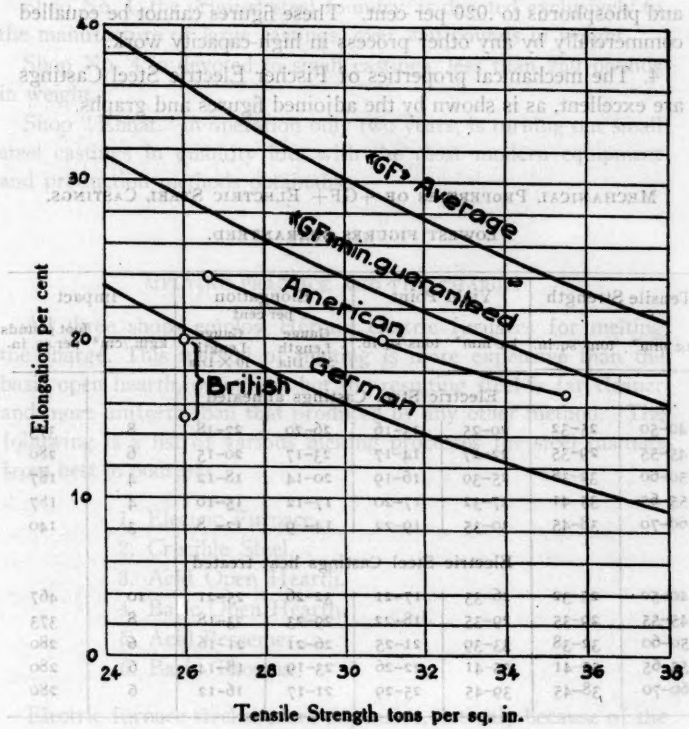


FIG. 2.

20-TON HEROULT FURNACE.

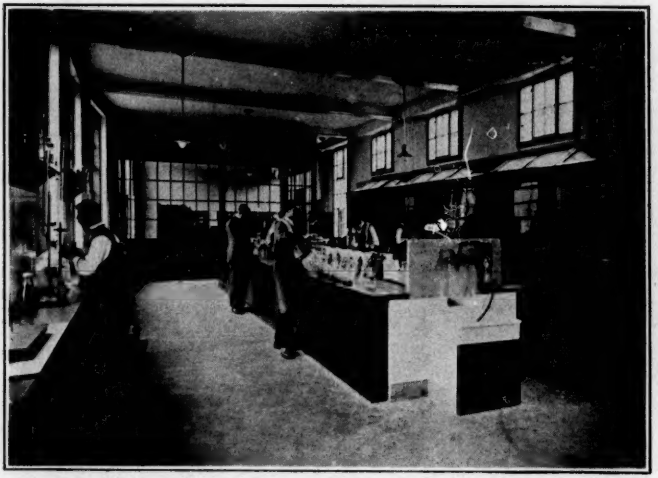


FIG. 3. CHEMICAL TESTING LABORATORY.

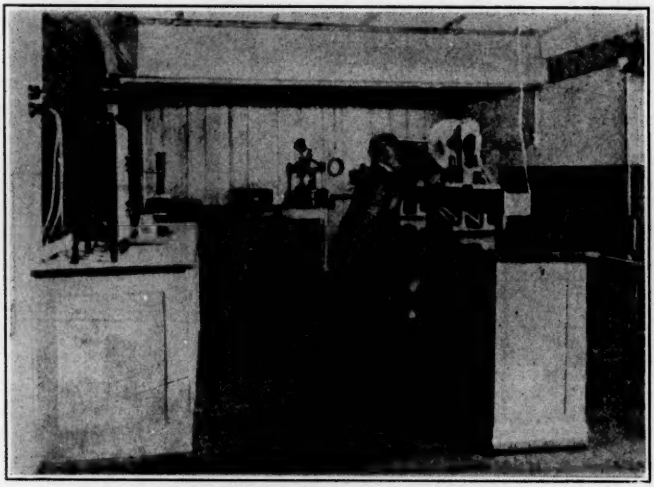


FIG. 4. PHYSICAL TESTING LABORATORY.

Fischer, therefore, employs exclusively electric furnace steel in all three of the foundries. The furnace equipment was stated in the plant to be as follows:

1. *Foundry No. 1.*

One 20-ton electric furnace for the largest castings. This furnace can only be run at night, during light load, from midnight to six a.m., because of the heavy power demand on the Schaffhausen Power Company.

One 10-ton electric furnace for routine work.

One 5-ton electric furnace to assist the larger furnaces.

2. *Foundry No. 4.*

One 10-ton electric furnace for all work.

One 5-ton electric furnace as reserve.

3. *Foundry "Ebnat."*

One 10-ton electric furnace for all work.

The furnace capacity mentioned above gives the highest charge. The furnaces are built in such a manner that the charge can be reduced to half of the maximum capacity without great commercial disadvantage.

Three graphite electrodes project through the roof of the furnace, each connected to one of the three phases of the low-voltage side of an electric transformer. The electric current arcs between each of the electrodes and the charge or bath. The furnaces are built for tilting.

FURNACE CHARGE.

The charge for the furnaces is very important since the final analysis is nearer the resultant of the various constituents of the charge than in any other process. The process is non-oxidizing during the last period of the heat.

As previously stated, the charge consists of (1) steel scrap (foreign) and (2) steel scrap (domestic).

(a) *Steel scrap (foreign).*

This material is the bulk of the charge and is probably the determinant of final quality. The greatest care must be exercised in its selection, for the electric furnace is peculiarly susceptible to poor scrap.

Steel scrap purchased by Fischer must have a history or pedigree. Fischer laboratories are said to know the final analysis of all the scrap purchased because of close cooperation with the plants furnishing her scrap. She is fortunate in having many high-grade machine works nearby, and these ship to Fischer their high-quality steel turnings, which comprise ninety per cent of the foreign scrap.

Fischer scrap is also purchased, whenever possible, from her own customers. This scrap consists, therefore, of turnings or "chips" from Fischer castings. In a sense, her steel is "inbred" as much as possible.

The foreign scrap is, consequently, the finest quality selected low-carbon steel turnings, or "chips," of which the pedigree is known to Fischer.

The writer observed that the scrap going into the furnaces was unusually "young," or clean and rust-free. The turnings were very long and curly, indicating a pure, mild, steel stock.

(b) *Steel scrap (domestic).*

"Domestic steel scrap" is the name applied to scrap drawn from the foundry itself. It consists, at Fischer, therefore, of refined electric steel chips, risers and turnings from the Fischer castings. It is hardly necessary to add that this is splendid material for the charge.

Domestic scrap is sorted into several grades, consisting of risers (lost heads) and turnings from the different grades of steel castings. The most important of these grades are (1) risers and turning from low-temperature normal castings and (2) those from high-temperature castings.

Large risers are broken up into smaller pieces in order not to interfere with the feeding of the electrodes. The top layer in the furnace consists of a dense bed of turnings at least 1 foot deep.

The charge, therefore, is all highly refined low-carbon steel. The final analysis of each constituent is known. The proportion of each is carefully controlled.

THE HEATING OF THE CHARGE.

According to the furnace capacity, the time for melting and finishing the charge is three to four hours. The temperature of the molten steel amounts to 2900 degrees–3100 degrees F. (Approximate only).

Some of the carbon in the charge is removed by the oxides of the slag. The melter pours samples, observes the fracture, and "ores down" if necessary to get the proper carbon content. This is done by throwing into the melt a carefully determined amount of iron ore and the oxides of the ore burn off the carbon in the form of CO_2. Care must be exercised not to reduce the carbon too low, otherwise the steel takes up too much gases.

The melter tries to "ore down" below the desired carbon content, but not below .05 per cent carbon.

"Oreing down" is not always necessary because of the low carbon charge and the action of the flux in removing carbon.

The furnace men then pour a small sample, break it in two, observe the fracture, send one piece to the Chemical Testing Laboratory, and the other half sample to the Physical Testing Laboratory.

It was stated in the laboratories that the furnace men are so skilled that they can, from the fracture, estimate carbon content invariably to one-tenth, and often to one-twentieth, of one per cent. The furnace man can estimate the physical properties to within five per cent of the laboratory determined figures.

SLAGS AND FLUXES.

The flux is added to collect the impurities. This slag consists of a great percentage of burned lime without any carbonate of lime. The test sheets showed a large amount of "Tonerde" or clay from the hearth or the roof of the furnace. After oreing down, the first slag is removed and the refining slag is put in the furnace. After the steel is completely deoxidized, the additions, as pig iron, ferrosilicon, ferromanganese, etc., are thrown into the melt. The ferrosilicon was said to contain about 50 per cent of silicon and the ferromanganese about 80 per cent of manganese.

After the additions are thrown in, test pieces are again poured for the final analysis and testing. The samples are broken and sent to the laboratories.

Summarizing, the materials employed for a normal melt in the 10-ton furnace are:

Charge

Domestic risers	4300 Kg.
Domestic scrap steel (turnings)	990 Kg.
Foreign scrap steel (turnings)	4440 Kg.
Total	9730 Kg.

Slag

first, oxidizing slag:	
burned lime	250 Kg.
iron ore	700 Kg.
second, deoxidizing slag:	
burned lime	100 Kg.
fluorspar	40 Kg.

Additions

Ferrosilicon, 50 per cent	113 Kg.
Ferromanganese, 80 per cent	70 Kg.
Pig iron	80 Kg.
Total charge exclusive of slag	10,000 Kg.

NOTE—The readings were taken at one of the so-called 10-ton furnaces.

POWER READINGS.

All electric furnaces at the Fischer steel works are of the high-power type. The power input is such that the charges in all furnaces are finished in three to four hours.

The power readings for the 10-ton furnaces, for example, are the following:

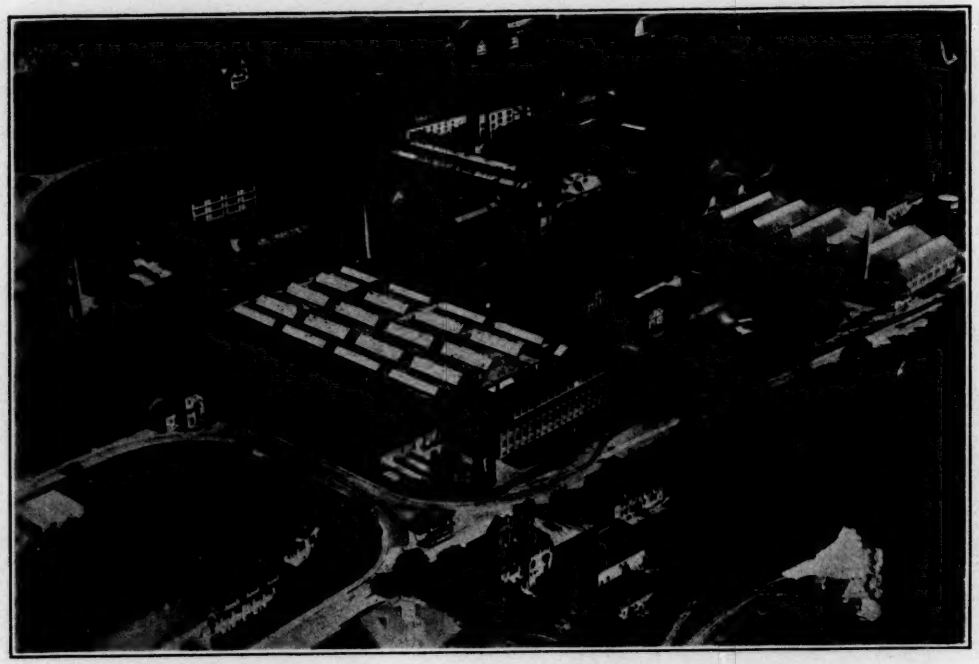
Transformer capacity, about 3000 Kw.

Melting-down period, 3000 Kw. 190 Volts.

Refining period, 1600 Kw. 110 Volts.

The Kwh-consumption is about 600-700 Kwh per ton.

GEORGE FISCHER CAST STEEL—SHOP NO. 4 FOR SMALL STEEL CASTINGS UNDER 200 LBS.



GEORGE FISCHER CAST STEEL—SHOP "EBNAT" FOR SMALL STEEL CASTINGS.

TAPPING.

The furnace is tapped at about 1600–1700 degrees Centrigrade. The temperature was determined merely by observing the melt by eye. It was stated that the furnacemen are entrusted to control the heating, mixing of the charge, and tapping without assistance from laboratories or pyrometers.

In the case of special alloys, however, the melting is controlled from the laboratories. The temperature determinations are always attempted with optical pyrometers.

The only thermocouple pyrometers seen were on the annealing furnaces.

FINAL ANALYSIS.

The following is given as a final analysis of the common Fischer Electric steel :

Carbon	.23
Manganese	.75
Silicon	.50
Phosphorus	.032
Sulphur	.030

To the above steel, many different alloying materials are added, depending on the uses to which the casting is to be put.

In more expensive and higher grade castings, phosphorus and sulphur are further reduced and nickel or other alloys are added.

Fischer stated that they did a very large business in steels for turbine castings, brake bands, etc., with a high creep limit. This steel was said to possess three times the physical properties of ordinary steel at 500 degrees Centigrade. Its composition was not disclosed.

PART IV.

MOLDING PRACTICE, POURING, AND THE AFTER TREATMENT.

Under the heading of "melting practice," the selection of materials for the charge, the composition of the charge, and the handling of the heat, have been briefly discussed.

Turning now to the materials used on the molding floor, and to the preparation of the mold, we find greater dissimilarity between the normal foundry practice and steel casting practice.

Since Shop No. 1 is making the large steel castings in which the Navy and Diesel manufacturers are chiefly interested, the following description will refer to this shop only in order to reduce the scope of this report. The other two shops, which are newer and use more up-to-date methods and equipment to perform production work, will be entirely omitted. The large Diesel casting is primarily "piecework." The manufacturer must rely on the individual skill of his labor to a far greater extent than is the case in quantity production. Here is the interesting field to the Diesel builder.

ANALYZING THE DESIGN.

Before going on to molding practice, one of the chief factors in Fischer success must be discussed; namely, cooperation with the manufacturer or customer.

When the plan of a new casting is received, it is always examined carefully by the Fischer technicians before being given to the pattern shop. Should this examination reveal a design which may be altered to assist the foundry without materially changing the design features or weight of the castings, the manufacturer is advised that such and such changes in the design will probably increase the chances of obtaining sound castings, increase the strength, or reduce the price.

The statement was made that the customers seldom fail to take this advice with the result that cheaper and sounder castings often result.

SAND AND MOLDING MATERIALS.

Of first importance in molding practice, is the selection of sand, and the placing of sand in the mold.

The statement was made that the use of sand at Fischer is the result of a century's experience with iron castings, and about forty years' experience with steel casting. Without great experience, the production of good thin section castings is impossible.

This may be greatly exaggerated. Nevertheless, the foundry which begins this type of work will have much difficulty and many rejections before learning the proper use of sand and steel molding.

It was also stated that this experience with sand must be taught the workman, even to the laborer shoveller on the molding floor.

At least a dozen different varieties of sand were seen in the bins of Shop No. 1. From these bins the sand is conveyed to conditioning rooms where it is prepared for the mixing rooms.

In the *mixing room*, each constituent of the mixture is controlled accurately by means of wooden forms subdivided into partitions, each one of which represents a known percentage of the whole sand form. By filling each partition with a different kind of sand, a mixture is obtained in which the percentage of each sand is known.

These *sand mixtures* are all standardized from knowledge obtained by experience. From the twelve or more sands employed, Fischer has evolved many formulas. The extreme care with which Fischer controls its sand mixtures appeared to be needless refinement in the opinion of several foundrymen consulted on the matter. However, the fact remains that Fischer have long been in the van in producing such castings.

From the mixing forms, the sand goes to the *sand mixers*, which consist of rollers spaced progressively closer together. The mixers knead the sand, break up the lumps, mix each ingredient thoroughly with the others, and produce a mechanical mixture rather than segregations of several varieties of sand. Each particle of sand should be of uniform size and rest in a matrix of clay binder. The sand particles should be at least 90 per cent of the total. It was stated that Fischer exercise unusual care in the mixing to produce:

a. Uniformity of result with each mixture.
b. Mold porosity for the dissipation of gas.
c. The proper binding to give mechanical strength.

The sand is taken from the "mixers" direct to the molding floor where it is used as facing, backing, or filling sand as the case may be.

Samples of some of the different sands and mixtures were obtained.

a. Greasy clay sand from the Pfalz or Palatinate, which is used as a binder and which was said to have excellent hardening properties after the molds are dried.

b. Krefeld and Duesseldorf fine silica sand.

c. Silica sand from the vicinity of Schaffhausen.

Molds were seen where as many as five different sands or sand mixtures were employed in one mold.

Ganister, or ground burned fire clay, called " chamotte" in Europe, was used by Fischer to a far greater extent than it is used in America. It was used in many of the mixtures; facing sands often appeared to consist of a very large percentage of ganister, and it was said to affect greatly:—

a. The mechanical strength of the mold to resist the erosion of the flowing metal.

b. The porosity of the mold to dissipate or vent the gas.

c. The rapidity of heat dissipation.

The *used sand* is cleaned, reconditioned, and sent back to the foundry floor to be used as filling sand. Shop No. 1 employs sand slingers for the filling. They break the sand up well, give a uniform pack, and aid in giving porosity to the mold.

The *graphite wash* (mold facing) with which the face of the mold is painted is a very important molding material. Fischer normally use an oil-graphite mixture of their own which was stated to give as good or better results than the many patented washes which Fischer has tried.

These washes make the face of the mold smooth and help to prevent erosion from the flowing metal. They should be non-gasifying, must go through the drying furnace or hot air blast without cracking, be mechanically strong enough at high temperature to withstand erosion, and at the same time give the proper venting.

Many European foundries, notably those of Luxembourg, employ *ground up used crucible pots* for this wash. It is probably the best material for the purpose, withal expensive and difficult to obtain. Fischer use this type of wash only for special work.

Summarizing:

(*a*) Fischer employ better grades of sand than can usually be obtained in America.

(*b*) They exercise extraordinary care in the mixing and preparation of the sand.

(*c*) Great attention is given to the selection of sand or mixtures for each face of the pattern.

(*d*) Fischer ascribes its success in steel casting very largely to her experience and knowledge of the mixing and use of sand.

Before leaving the subject of sand, mention should be made of the *cores*. These are oil and sand mixtures baked after application of a core wash similar to the ground coke or graphite used in the mold proper.

Both the core wash and the binder must permit venting, and Fischer use a large grained sand which has been ground for hours in a special machine similar to the mixing machine. It was stated that linseed oil is used for the binder.

THE PREPARATION OF THE MOLD.

Flasks are universally used at Fischer in Shop No. 1, although Sulzer do their casting in the ground at Winterthur near by. In Shop No. 4 where smaller castings are made, many of the very thin sectioned and complicated castings were made with flaskless molds. In Shop No. 1, however, all molding, even for locomotive frames and large turbine castings, was done in flasks.

The *facing sand* and *backing sand* is built up around the pattern by hand. Much care is exercised in selecting the proper sands for the casting sections, especially at the corners. Here much clay binder is in the mixture to prevent erosion.

After the mold has been faced, the *sand slinger* is used to fill it up. *Shadows* in the mold, resulting from the pattern blanketing certain sections from the blast of the slinger, are tamped by hand with *air rams*.

The placing of *risers* is one of the most important factors in successful casting. The shrinkage in steel casting is so great and the steel is in a soft, spongy, tough state for so long, that these reservoirs of heat and molten metal are most necessary to prevent cracks, blow holes, and bad flaws. The risers are often 150 per cent of the useful casting weight.

Again, the locating of these risers is the result of both the experience and skill of the workmen. The heads must be placed where

they will prevent the hardening of the thin, fast cooling sections, before the heavy slow cooling sections, still in the molten state, are strong enough to resist the tendency of the contracting thin section to pull away.

Risers were observed running from a common point at different angles. Some large risers had smaller *auxiliary risers* leading from them at an angle. The purpose of this practice is obscure.

The mold is fitted with many *gaggers,* or heavy iron reinforcements which prevent the sand from falling and hold it together during the handling of the mold on the foundry floor. Very large molds were fitted with *strongbacks* from the flask running through a hock in the gaggers.

Mold nails are small headed iron nails, the heads of which are treated with tin and dipped in graphite wash. They are pushed into the sand after the foundryman prepares a hole for the head about ¼ inch deep with a trowel. This hole is then rubbed over with graphite. The nail is therefore slightly countersunk. Some castings have innumerable mold nails all put in by hand.

When mold nails are used as chills, however, the heads are not countersunk. Such castings often have hundreds of these nails projecting from them. This form of chilling is resorted to only when necessary and when the surface is later to be machined.

Chills or *coquilles* are made of tin covered cast iron and are used to chill heavy sections to give the same rate of cooling as adjacent thin sections. They also sometimes prevent the erosion of sand. The tin covering prevents oxidation of iron and the possible generation of the gas.

Chaplets are only used when absolutely necessary.

It was stated that chills and chaplets are used far too much by steel casting foundries, and that they should be avoided whenever possible.

The *drying* of the mold is very important to prevent gas generation from entrained moisture. Shop No. 4 dries in furnaces, but Shop No. 1, making large castings, is not so equipped. The drying is performed by blowing hot air through the mold. This is done by building a fire in the mold and running an air blast through this fire.

TAPPING OFF AND POURING THE MOLDS.

The melt from the furnace is tapped in " stopper"-ladles, from which the steel is poured through the ladle bottom into the molds. The advantage of the stopper ladle is to prevent the slag from running into the mold.

Therefore the poured steel is absolutely clean and free from slags.

The ladles are handled by cranes, and the time of transport, especially in thin section work, is reduced to an absolute minimum to prevent the cooling of the steel in the ladle. Several ladles are sometimes used on one large casting. As many as five gates are occasionally employed. With multiple gates, extreme care must be exercised to time the pouring in order to prevent "cold shuts" in the thinner sections.

The molten steel is poured until every riser nearly overflows. The foundry laborers then, immediately, shovel a very heavy layer of sand over all risers to hold the heat in, thus preventing cooling of the risers by the air. This keeps the riser molten while the slag rises from the mold. It also permits the molten metal of the riser to feed the shrinkage as long as possible.

CLEANING THE CASTINGS.

Very thin section castings are allowed to cool in the mold much longer than normal practice. Molds were observed being shaken, and in which the metal could be touched by the hand.

After shaking out, the molds are cleaned by air chisels, air hammers, and air brushes. They are then sand blasted if necessary.

Following the cleaning of the castings, the risers or lost heads are cut off and the casting is inspected for flaws.

ANNEALING.

The castings are annealed in order to refine the very coarse grains which are characteristic of metals very slowly cooled through the solidus. Annealing also is done in order to remove the strains due, not only to uneven cooling, but also those due to the large contraction of cast steel. The castings are heated slowly in an annealing furnace to about 850 degrees Centigrade. Shields are

often used and the temperature is controlled by means of thermo-couple pyrometers. Large castings are annealed for at least 24 hours.

INSPECTION AND MACHINING.

Following annealing, the castings are again carefully inspected. Some defects may be repaired by welding if they are located where the casting's strength and utility will not be affected. Most of the welding outfits were oxy-acetylene. Shop No. 1 had an electric welding outfit for the repair of large castings.

The castings are then taken to the shop for rough machining. Shop No. 1 has a complete machine shop fitted with large machines to rough out its castings.

After the castings are machined, they are again very carefully inspected for flaws. They are also sometimes reannealed.

PART V.

THE LABORATORIES.

The Chemical and Physical Testing Laboratories exist in order to *control* the foundries. Much of the *research* work is done by independent metallurgists retained by the company.

The laboratories are plentifully supplied with money and equipment.

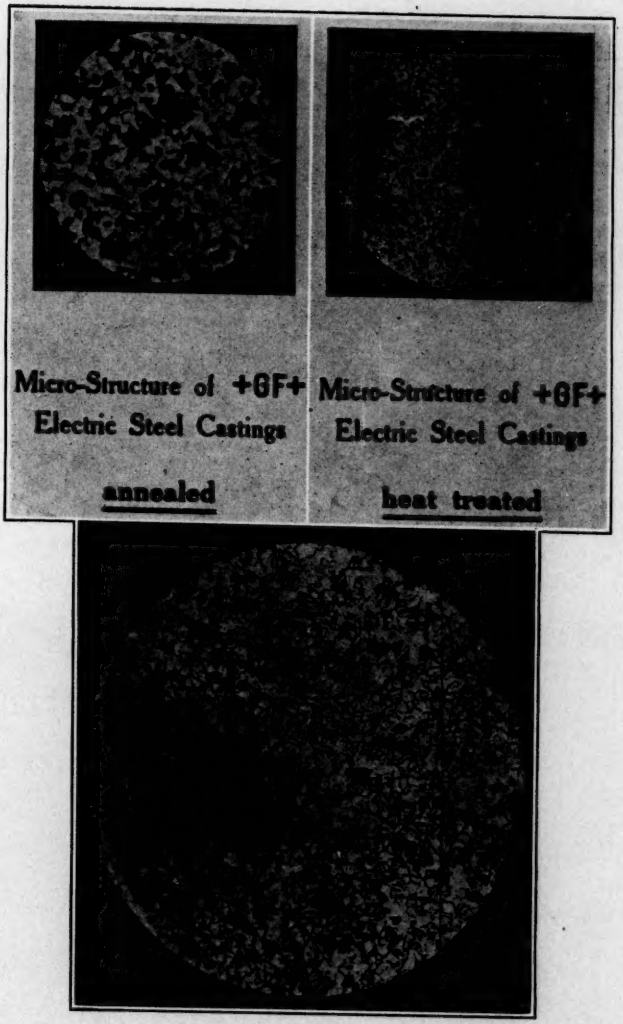
CHEMICAL TESTING LABORATORY.

In the Chemical Laboratory, samples of every melt are tested. It receives half of the samples referred to in Part III. These are drilled to give the necessary chips for a *quantitative* test.

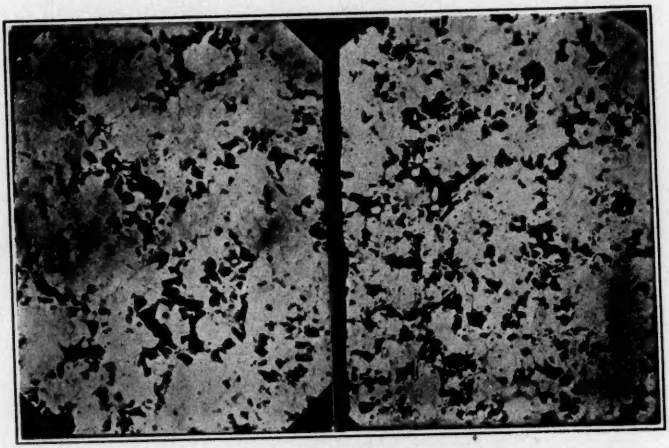
The fractures are also carefully examined to check the furnace man. A complete copy of the report is returned to the foundry for the information of the furnace men.

Until last year, Fischer had filed the reports and samples of every melt made in their furnaces since the founding of the company. Last year, however, the accumulation became so great that the samples and reports dated before 1920 were placed in dead storage.

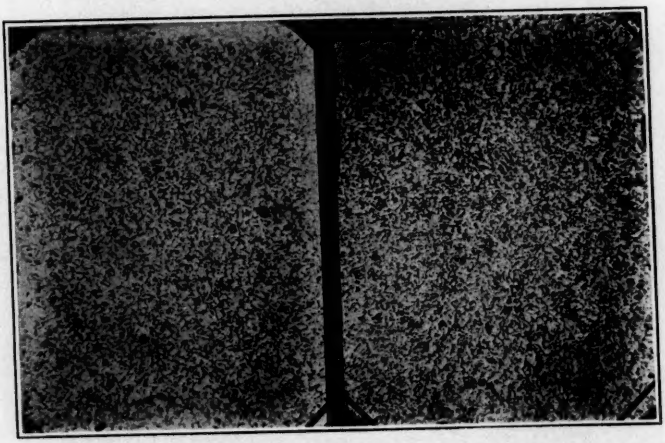
These samples are referred to if a complaint is received on a casting from some particular melt. Every casting is stamped with the serial number of the furnace charge from which it was cast.



FIGS. 5, 6, 7. MICROPHOTOGRAPHS FROM FISCHER LABORATORY.



FIGS. 8 AND 9. NO. 933 ANNEALED.



FIGS. 10 AND 11. NO. 934. HEAT TREATED. × 100.

PHYSICAL TESTING LABORATORY.

The Physical Laboratory, like the Chemical, is used to control the foundries and give Fischer complete information on all castings manufactured.

The majority of the equipment was the work of Amsler, located near by in Schaffhausen. One new type of hardness testing machine, however, was observed to be of American manufacture.

In order to test for tensile strength, the high temperature steel used in brake bands, turbine casings, etc., the samples were placed between the jaws of the machine surrounded by a thermostatically controlled electric furnace.

The physical testing laboratory used a Zeiss microphotographic outfit.

Great sheets were observed showing the physical properties of many thousands of charges plotted graphically. Each standard alloy was plotted on its separate sheet.

Upon these graphs Fischer bases guaranties. For instance, one particular alloy was pointed out which is guaranteed to give 20 per cent elongation, and less than a dozen below showed 25 per cent. Mr. Hübscher stated that Fischer may now guarantee 25 per cent instead of 20 per cent for this particular steel. However, since 20 per cent is a better figure than competing firms will guarantee, Fischer is holding this 5 per cent "up its sleeve" until she is faced with serious competition.

PART VI.

COMMENTS OF AMERICAN AUTHORITIES.

In order to obtain the comments of a well-known authority on the metallography of the Fischer work, samples of annealed and unannealed steel were sent to Dr. William Campbell of Columbia University. The comments of Dr. Campbell are as follows:

TWO STEEL CASTINGS—ANNEALED AND HEAT TESTED

No. 933 & 934.

Two samples of Steel Castings were received from Lieut. John Huse, U. S. N., office of the Naval Attache, Berlin, for examination. They were in the form of broken Charpy test pieces, and were marked as follows:

No. 933 Annealed

No. 934 Heat treated

Cross and longitudinal sections were cut and polished then examined for non-metallic inclusions. To develop the structure, these specimens were etched with picric acid in alcohol and type micrographs taken at a magnification of 100 diameters.

No. 933—Annealed. As polished shows numerous small non-metallic inclusions, in amount similar to those in Acid Open Hearth Steel. On etching, the metal shows a comparatively fine grain, is of low carbon and the pearlite not very regularly distributed. It resembles the Nickel Steel Castings from the Washington Navy Yard (See "On Heat Treatment of Steel, Part I, Section B, Steel Castings, Grades D.A.E.&F," by Wm. Campbell, D.Sc., Metallurgist, Navy Yard, New York, Feb. 10, 1920).

Figs. 8 and 9 show the average structure. The small, dark areas of pearlite are somewhat irregular in their distribution, while the matrix of ferrite has a fine grain. The small, non-metallic inclusions can be seen in the ferrite. They are comparatively small and fairly evenly distributed, and as such, have but little effect on the metal's strength and ductility.

This sample as polished shows non-metallic impurities in about the same amount as before, and these are comparatively harmless also.

On etching, a marked difference in structure is seen. The grain is very fine and the dark etching areas are well distributed. They might be called "granular pearlite," but under a high power they resemble the precipitated carbide particles obtained by quenching low carbon steel from Ac_3 and reheating to a dull red, say 600 degrees C.

Figures 10 and 11 are typical of the structure of this specimen.

Such a fine structure would be difficult to obtain in a plain low carbon steel and hence it is probable that this is a nickel or other alloy steel.

WM. CAMPBELL,
Howe Professor of Metallurgy,
Columbia University.

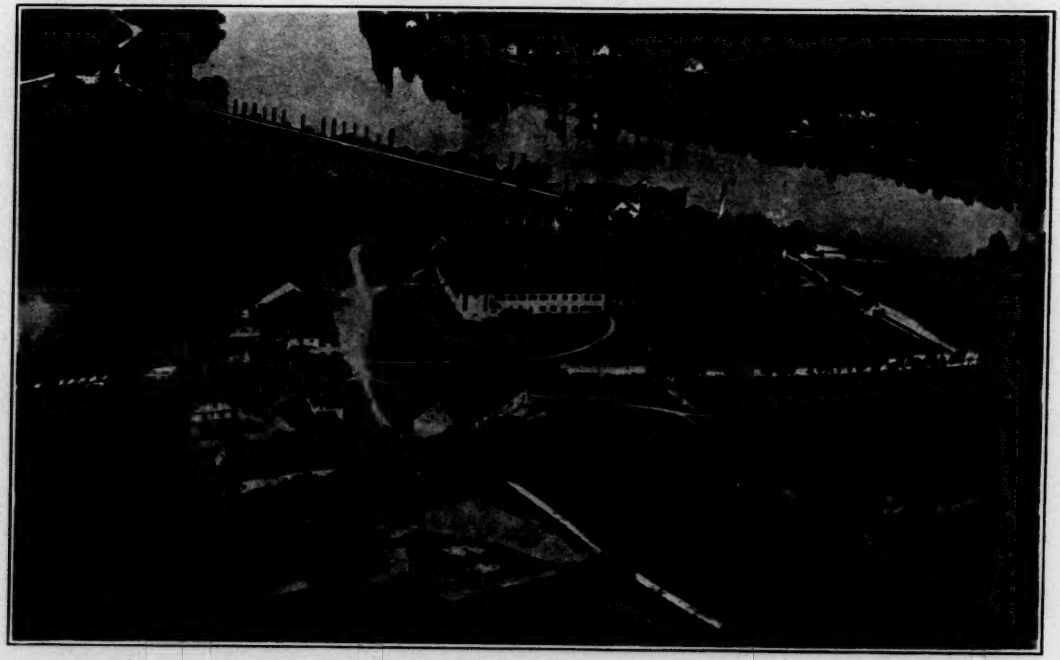
July 8, 1930.

PART VII.

CONCLUSIONS REGARDING FISCHER PRACTICE.

An attempt to point out, from these incomplete observations, all the reasons for Fischer world supremacy in the difficult art of steel castings would be presumptuous. However, the following contributory reasons are patent:

1. The *long experience* of Fischer is, perhaps, second to no other steel foundry in the world. This statement is certainly true of lightweight, thin section castings for high speed Diesel engines.



GEORGE FISCHER CAST STEEL—WORKMEN'S SETTLEMENT.

GEORGE FISCHER PLANT—MALLEABLE IRON FITTINGS.

2. Fischer *labor is highly trained,* stable, and relatively well paid; far more hand labor, care, and skill goes into the casting than in America.

3. Fischer employ the most *expensive practice* throughout if this expenditure of money will improve the quality of the casting. In other words, they adopted all the finest practice possible and cut costs only by training or employing skillful labor to reduce the percentage of rejections.

4. Fischer select only the *best materials* for the melt.

5. *Better sands* are available in Schaffhausen than are usually available in the United States. Special emphasis has been placed on the selection of sands, the mixing of sands, and the use of these mixtures to aid in improving the properties of the mold which overcome the peculiar difficulties met in the steel foundry.

6. The *close cooperation between Fischer and her customers* makes it possible to avoid difficult feats of founding where they are unnecessary.

The last factor, namely, that of cooperation, is believed to be of the utmost importance. The statement was made that the cost is often halved and a difficult job made easy by a small change in the plan having no material effect on the design nor upon casting weight.

Fischer stated that the standard minimum wall thickness in large lightweight castings is six millimeters, or slightly less than one-quarter inch. This figure is known to all her customers. Lighter sections are never demanded except in special cases, and for good reasons. Eight millimeters is the standard for commercial castings where weight is not of first importance.

The limited foundry experience of the correspondent has made it most difficult to locate in detail the factors which give Fischer the predominance in the steel casting field.

In conclusion, therefore, no stronger emphasis may be placed on the fact of Fischer leadership in the art than to use the words of a practical man who is the eminent head foundryman of one of our largest corporations, and to use the words of one of our leading scientific metallurgists.

The former stated, through his representative, "He was more impressed by the work of Fischer-Schaffhausen than any other steel

foundry visited." This statement followed an inspection trip which included all the steel foundries of Europe.

The latter, Dr. Campbell, writes:

"Answering your question, I don't understand how the Navy can afford not to use such a grade of steel in submarine work. Three per cent nickel is not so expensive and modern heat treatment is cheap. The Army use such material and so does Ordnance." Furthermore, Dr. Campbell states that the failure to use such material is "to save at the spigot while it leaks at the bung."

COMBUSTION IN THE FURNACES OF MARINE BOILERS.*

By T. B. STILLMAN,† MEMBER.

INTRODUCTION.

Because of the limitations of space and weight prevailing in marine work, the generation of power on shipboard is limited to the two fuels—oil and coal. Wood, gas, bagasse, coke, etc., which are used extensively in different parts of the world for the generation of power on shore are eliminated from marine practice because of their large volume for a given weight and B.T.U. value, an item of great importance in the case of any fuel used at sea. It is the fact that oil is much more compact for a given B.T.U. value than any other commercial fuel, that makes it so desirable aboard ship, especially in the case of passenger vessels where space is at such a premium. The fact that oil is liquid at ordinary temperatures and may be readily pumped from one part of the ship to another is also an item of importance in its popularity for marine use, and were it not for the cost incurred by its use at times, this paper would be devoted exclusively to the combustion of *oil* in the furnaces of marine boilers.

Although from the chemical aspect, the combustion of coal and oil are very similar, consisting in both cases of a combination of hydrogen and carbon with oxygen, the methods employed in securing this combustion are so different with the two fuels that they will be treated separately, a number of the more usual methods used with each fuel being given.

OIL.

A. *Air Atomizers:*

When oil was first introduced into marine work for power generation purposes, air atomizers were sometimes used in salt

* Paper presented at the World Engineering Congress, Tokio, Japan, October, 1929.
† Engineering Dept., The Babcock & Wilcox Co., New York.

water ships, as in this way a very fine atomization was readily secured without the loss of any fresh water. It was soon learned, however, that the cost of compressing the air for atomization was very high, and also that the weight of the compressors was an undesirable feature for marine work. Roughly, it costs eight times as much to atomize a thousand gallons of oil with compressed air as it does with a modern pressure atomizing system. This combination of excessive weight and high cost of operation of the air atomizing type of oil burners gradually reduced their use in the marine field until they are practically unknown there at the present time.

B. *Steam Atomizers:*

As in the case of the air atomizers, these were carried into the marine power plant field when oil was first introduced there, because of the knowledge that had been gained of their use in stationary installations. They had one serious drawback for salt water ships, and that was the loss of fresh water entailed from their use with the result that they were never used extensively except on fresh water, and for short runs (around harbors, etc.) on salt water.

The flat type of flame which was usually employed in conjunction with a checkerwork of firebrick for admission of the air to the furnace was limited in capacity in the small furnaces available under marine boilers, any attempt to force this combustion to high rates resulting in excessive smoke and high hydro-carbon and other combustible losses. These two features, loss of fresh water and limited capacity soon began to eliminate the steam atomizers from marine work, until at the present time they are never seen except in a few old installations on harbors or rivers. Steam atomizers never had anything to recommend them for marine work except their low first cost, and that saving was so trifling compared to the losses their use involved that in modern marine work they are never employed.

C. *Pressure or Mechanical Atomizers:*

Oil burners using this type of atomizer are almost universally installed in modern oil burning ships. Although there is a great number of this type of burner made throughout the world, in gen-

eral principle they are all alike. An atomizer located in the center of an air register, discharges oil into the furnace in the form of a hollow conical spray, which is produced by the centrifugal action of the oil under pressure passing through suitable passages located near the tip of the atomizer. The air for combustion is brought to the register so as to surround and mix with the conical oil spray to produce the combustion desired. Various designs of tips are employed to produce the hollow conical spray, and the fineness and angularity of the cone of this spray varies accordingly in different types. In a similar manner, the different designs of air registers use different means of directing the air so as to have it mix with the oil spray, some producing a very rapid and complete mixture; some a delayed mixing with slow burning flame, and some never giving a complete mixing at any time.

It is of course essential that the oil, when it reaches the tip of the atomizer, be sufficiently fluid so that it will readily break up into a spray, due to centrifugal action, as soon as it is released from the orifice. This requires that its viscosity be reduced to at least 3 to 5 degrees Engler, or 100 to 120 degrees Saybolt. In general, it may be said that the wider the angle of oil spray, the finer the atomization and the greater the speed of combustion will be. It should not be inferred from this that wide angle sprays are always the best to use, even in the small furnaces available with marine boilers.

In the case of Scotch boiler furnaces it is impossible to use very wide angles because of the tendency of sprays of this type to build rings of carbon in the furnaces, these rapidly growing to a point where it is necessary to shut down the burner. Fortunately, the length of travel available for the flame in the furnace and combustion chamber before the tubes are reached in this type of boiler is sufficient, so that with the relatively small amounts of oil burned in these furnaces, fairly satisfactory results may be obtained with the narrower angles of spray, provided a well designed air register is used. Because of the relatively low furnace temperatures existing in Scotch boilers, it is very easy for a poorly designed burner (combination of atomizer and air register) to cause big fuel losses, either due to the necessity of using

large amounts of excess air to prevent smoke, or, if large amounts of excess air are not used, still greater losses due to smoke and unconsumed hydro-carbons passing up the stack.

In furnaces under water tube boilers of the straight tube, header type, where the tubes are fairly long and reasonable travel is given to the gases before they enter the tube bank, moderately wide angles of oil spray are desirable, so that the flame reaches the full length of the furnace. Care should be taken not to have the angles of spray too narrow in furnaces of this type, or the combustion will be slowed down to the point where it will be incomplete before the tubes are reached, and, conversely, the very wide angles are to be avoided, as, if the combustion is too rapid and concentrated near the front wall, maximum use is not made of the radiation effect from the flame into the furnace row of tubes. Unless correct use is made of the furnace radiation by means of proper flame distribution in the furnaces of this type, a distinct loss in boiler efficiency will result.

In the case of water tube boilers of the three drum Express type where it is necessary for the flames to pass directly to the tube bank from burners located close to it, wide angle sprays are absolutely essential for best results. Also, as it is usual to operate this type of boiler at higher rates (especially in Naval installations) than the heavier types of boilers, the speed of combustion becomes a very important factor to prevent smoke and unconsumed hydro-carbon losses, and the widest angles of oil spray combined with the proper types of registers are necessary to give the desired results.

From the above it may be noted that no one size or design of burner is best for use in all types of marine furnaces. It is only by careful experimental investigation that the best combination of oil sprays and registers can be developed to meet the widely varying furnace conditions encountered in marine practice, and the different cases as they arise should be handled accordingly, if the efficient results sought are to be obtained.

Tables No. 1, 2 and 3 give representative boiler efficiency tests which show what may be expected with properly designed and efficiently operated oil burners used under Scotch, Babcock & Wil-

cox Marine and Babcock & Wilcox Express type boilers respectively. None of the boilers with which these tests were run was fitted with air heaters or economizers. The water and oil in each case was carefully weighed on calibrated scales, and the readings throughout were taken with the greatest accuracy. In Plate No. 1 the three upper curves show the efficiencies and the three lower curves the total radiation and unaccounted for losses of the heat balances of the tests shown on Tables No. 1, 2 and 3. The total radiation and unaccounted for losses were obtained by deducting the sum of all the accounted for losses shown in the heat balances plus the boiler efficiencies from 100 per cent.

In studying these latter curves it is interesting to note the form these losses take under different conditions and at different rates of operation. Per cent radiation by itself is always greatest at the lower rates, and lowest at the higher rates of operation of the boiler, whereas the hydro-carbon and other combustible losses usually increase with rating. The curve at the bottom of Plate No. 1 of The Babcock & Wilcox Express boiler from the 4.0 pounds to the 12.0 pounds of water per square foot rate represents almost pure radiation with a very small hydro-carbon or other combustible loss up the stack. From that point on the curve turns upward, due to the combustible losses increasing at the higher rates to a point where their increase is appreciably greater than the corresponding reduction in radiation. The Scotch boiler curve shows this same characteristic to a marked degree at the relatively low rates of operation. If the reduction in radiation is about counterbalanced by the "unaccounted for" losses, a horizontal line for the total of these losses will be secured over practically the entire range of operation of the boiler, as shown in The Babcock & Wilcox curve at the bottom of Plate No. 1. In addition to the radiation and unconsumed combustible losses mentioned above, there are other slight "unaccounted for" losses from boilers, which vary with different installations and which cannot be accurately segregated, such as heated air escaping from the fireroom and air leakage through the boiler casing, this latter seldom being properly determined by the flue gas analysis, and hence not being properly accounted for in the heat balance.

COAL.

A. *Hand-fired:*

In the marine field practically all coal used is hand-fired, the exceptions to this in actual service being trifling in number and relatively negligible. However, there has lately been a marked tendency shown by up-to-date ship operators to start installing apparatus which will handle this fuel more efficiently, and without the heavy manual labor required for hand-firing. As time goes on the practice of hand-firing will undoubtedly grow less and less as the more modern methods are introduced, and there will be as little chance of a new ship being built to operate hand-fired, as there is at the present time for a modern stationary steam power plant to be so equipped.

The efficiency results that may be expected with hand-firing are so dependent upon the firemen and of the grade of coal used that definite tests are of little use in making anything but general statements of what may be expected with a given piece of apparatus. Changing firemen may make a difference of 10 per cent or more in the efficiency results secured with a given boiler, and the same may also be said for a radical change in the coal. A great many hand-fired tests are on record with which reasonable care was taken to insure proper attention to the fires and the coal used well adapted to this method of firing. From these an approximate generalization may be made as follows—

Scotch boilers without air heaters—60 to 65 per cent efficiency;

Scotch boilers with air heaters—70 to 75 per cent efficiency;

Water tube boilers without air heaters—65 to 70 per cent efficiency;

Water tube boilers with air heaters—75 to 80 per cent efficiency.

It is realized in making these general statements that there are many installations in service that fall wide of the figures given, but in the average ship, they should be approached, with proper attention to the firing, reasonably suitable coal, and properly designed and clean boilers. It is of interest to note that the air heaters are responsible for an increase of approximately 10 per cent in the average efficiency figures, even though there are practically no air heaters in service in marine work that would give

this increase because of the lowering of the temperature of the uptake gases alone. It is necessary to cool the uptake gases coming from the average hand-fired boiler approximately 35 degrees F. for each 1 per cent gain in boiler efficiency, and the average air heater in marine service does not cool these much more than enough to account for about 5 per cent in efficiency. The remainder of the gain shown is obtained by improving the combustion conditions in the furnaces due to the use of the pre-heated air, which has the effect of reducing both the carbon loss in the ashpit and the unconsumed carbon and hydro-carbon losses up the stack, these losses being very much in evidence in the average hand-fired installation. The speed of combustion necessary in the relatively small furnaces available in marine work, again is the important factor, the preheated air assisting materially in speeding up the combustion, increasing the furnace temperature, and reducing appreciably the combustible losses which occur without its use.

B. *Pulverized Coal:*

During the last year or two a good deal of attention has been given to the use of pulverized coal on shipboard as one of the means of eliminating the labor of hand-firing and improving the efficiency of the ship. A few installations have been made and are in service, and more are being put in at the present time. Although this method of firing coal under boilers has experienced a very rapidly increasing use in the stationary power plant field for some time, its application to the small furnaces available in marine work was not successfully accomplished until lately.

In the first attempts to do this, pulverized coal burners of the "stream line" type, similar to those utilized in the majority of stationary installations, were tried, and the results secured impractical for commercial installations. The loss of unconsumed carbon up the stack was so great that efficiency results as low as those obtained by hand-firing on the same boiler were secured, and trouble was also experienced from excessive slag formation caused by the delayed combustion conditions existing. It was not until burners using registers giving a turbulent action to the air, similar to those used with the more efficient designs of mechanical atomizing oil burners, were employed, that the combustion

was speeded up sufficiently to give satisfactory results. Also the "stream line" of coal discharged into the furnace had to be broken up, so the particles of coal were enabled to mix thoroughly *at the burners* with the turbulent entering air. With this combination of a properly distributed supply of coal and a turbulent mixture of the coal with the secondary air, satisfactory results were secured and commercial installations on shipboard became possible. Tables No. 4 and 5 give test data showing the efficiency results secured with turbulent types of pulverized coal burners on a Scotch boiler and a Babcock & Wilcox marine boiler, respectively. The Scotch boiler was the same one used for the tests reported in Table No. 1, and the data are reproduced herewith from a paper by Mr. Carl J. Jefferson, presented before the 35th Annual Meeting of the Society of Naval Architects and Marine Engineers in November, 1927.

It may be noted that for the pulverized coal tests this Scotch boiler was fitted with an air heater, a very desirable adjunct for speeding up the combustion to a satisfactory extent with this type of fuel in these relatively cold furnaces and also serving to preheat the supply of primary air going to the mill. The Babcock & Wilcox boiler used for these pulverized coal tests was the same as that used in the tests reported in Table No. 2, no air heater being used with this boiler for either the oil or pulverized coal tests. The furnace used for pulverized coal under this boiler is shown in Plate No. 2. A small steam air heater was utilized to preheat the supply of primary air to the mill for the pulverized coal tests run with this boiler. The two upper curves of Plate No. 3 show graphically the efficiencies reported in Tables No. 4 and No. 5 and the two lower curves, corresponding total radiation and unaccounted for losses.

To date it has not been practical to use pulverized coal with small tube Express type boilers because of slag filling the small gaps existing between the tubes in this type of boiler and preventing the escape of the gases from the furnace. The problem with this type of boiler is also complicated by the fact that most of these boilers in service are operated at rates considerably in excess of 100,000 B.T.U. release per cubic foot of furnace volume per hour, and at these rates the carbon losses up the stack would

be excessive, even if the slagging of the tubes did not prevent operation.

Pulverized coal is a much more difficult fuel with which to obtain relatively complete combustion in the small furnaces available in marine work than is the case with oil. The more the furnaces are cooled, the more difficult it becomes, the losses increasing rapidly with increased rates of B.T.U. released per cubic foot of furnace volume. The curves shown on Plate No. 3 clearly emphasize this point, the Scotch boiler furnaces being completely water cooled except for a distance of about 2 feet near the burners where a cylindrical refractory lining was used. The furnace under the Babcock & Wilcox boiler was lined with insulated brick throughout except for the furnace row of boiler tubes. It may be noted by comparing the data in Tables No. 4 and No. 5 that when releasing 60,000 B.T.U. per cubic foot in the Babcock & Wilcox brick lined furnace, the total radiation and unaccounted for losses were approximately 6 per cent, and when releasing 60,000 B.T.U. per cubic foot in the Scotch boiler furnaces this loss rose to approximately 16 per cent. This in spite of the fact that the pulverization of the coal used at this rate was a little finer with the Scotch boiler than with the water tube type. As may be noted, the same burners were not used for the Scotch boiler tests and the Babcock & Wilcox boiler tests.

Following the completion of the brick lined furnace tests with the Babcock & Wilcox boiler, 15 square feet of water cooled surface was used in each side wall, this resulting in an increase of nearly 2 per cent in the unaccounted for losses at the 60,000 B.T.U. per cubic foot rate. At low rates of B.T.U. released per cubic foot of furnace volume (about 30,000 B.T.U. per cubic foot or less), there was no loss in boiler efficiency, due to the use of the increased water cooled surface in the furnace, the increased heat absorption of this surface counterbalancing the slight increase in carbon loss. These rates are too low, however, to be considered for practical operation in marine installations.

From the above it may be noted that the higher the furnace temperatures used with pulverized coal in small marine furnaces, the more complete the combustion at a given rate of operation. Care must be taken in these small furnaces to keep the fusing

temperature of the ash in the coal above the furnace temperatures existing near the walls and tubes, or slagging will cause operating difficulties. Also the pulverization of the coal should be as fine as it is commercially possible to have it, as the finer the pulverization of the coal, the lower the carbon losses, and the less the slag trouble in the furnace when operating at a given rate with a given coal.

To sum the matter up, to secure the best results with pulverized coal on shipboard, careful investigation should be made of the particular installation under consideration, to insure the best compromise between furnace temperatures, carbon losses, fineness of pulverization, slag difficulties, and boiler efficiencies. All of these must receive due consideration if the advantages to be expected from the use of pulverized coal on shipboard are to be realized to their fullest extent.

C. *Stokers:*

The application of stokers to marine boilers is not new, but it was only during the last few years that their use may be said to have become commercially desirable. Previously the relatively low price of fuel and labor counteracted to a considerable extent the expense of installing stokers, their weight, and the operating difficulties of some of the earlier designs. This picture is now radically changed. The price of coal in the bunkers is higher than it was before the war, and the difficulty of securing competent firemen who will remain with a ship that is hand-fired is becoming a real problem. Also the experience gained in the stationary field in building reliable stokers is of the greatest value when applied to marine practice, where reliability is absolutely essential. A number of ships are now in regular service fitted with modern stokers, and the results obtained fully justify their use.

The principal application of stokers in the marine field has been under water tube boilers, their application to the Scotch boiler not having been commercially successful; the shape of the furnace and the amount of cooling surface present in a furnace of this type of boiler being primarily responsible. Where high furnace settings are available under water tube boilers, the Underfeed type of stoker may be used with good results, especially if

there is room below the stoker for a deep ashpit and clinker grinder. For the low set water tube boilers, such as are usually encountered in marine work, the Chain Grate type of stoker is undoubtedly the most suitable. The design is such that a refractory arch may be used over the coal bed, insuring high furnace temperatures, and a correspondingly rapid, efficient combustion, so necessary for these small furnaces. By using a chain grate stoker of the Forced Blast type, with the air supplied from a number of separately controlled compartments under the grate, excellent control of combustion is secured and correspondingly high boiler efficiencies are obtained. The power required to operate this type of stoker is small, an item of importance on shipboard, where auxiliary power is usually generated in small and relatively inefficient apparatus.

Plate No. 4 shows a Babcock & Wilcox Forced Blast Chain Grate stoker installed under a Marine Water Tube boiler of that make, used for test purposes in the Bayonne, N. J., plant of that company. This is the same boiler that was used for the oil burning tests reported in Table No. 2 and the Pulverized Coal tests reported in Table No. 5. The standard setting height was not changed, this leaving a furnace volume of 167 cubic feet above the stoker. An ash conveyor manufactured and supplied by the Allen-Sherman-Hoff Company of Philadelphia, Pa., was installed at the discharge end of the stoker to crush and move the ashes from the stoker to a convenient spot for disposal.

Table No. 6 shows the evaporation results secured with this Forced Blast Chain Grate stoker, and Plate No. 5 shows graphically the efficiencies and total radiation and unaccounted for losses recorded in Table No. 6. From an analysis of these results it may be noted that even at very high rates of B.T.U. released per cubic foot of furnace volume, excellent combustion results were obtained with high CO_2 values and negligible smoke in the products of combustion.

A number of different coals were tried on this stoker, including high and low fusing temperature ash coals and high and low volatile coals. Excellent operating results were obtained with all of these coals, although the boiler efficiencies obtained with them varied, depending upon the coking properties of the coal, and the

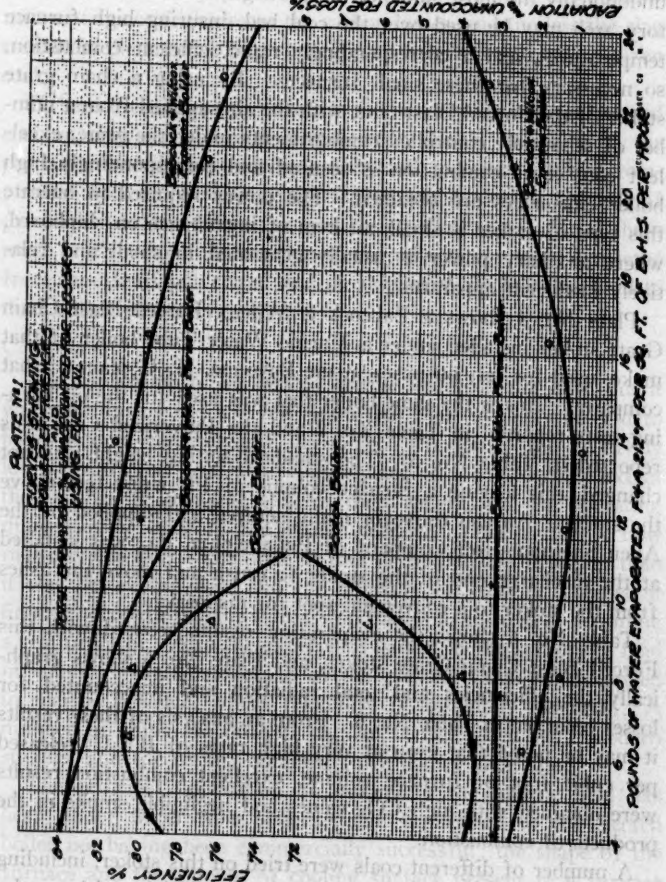


PLATE N°1
CURVES SHOWING
POWER EVAPORATIVE
EFFICIENCY
AND
EVAPORATION
WITH
FORCED DRAUGHT
AND
NATURAL DRAUGHT
WHEN BURNING
FUEL OIL
Forced Draught
Natural Draught
Forced Draught
Natural Draught
Forced Draught
Natural Draught
EFFICIENCY %
EVAPORATION % UNACCOUNTED FOR LOSS %
POUNDS OF WATER EVAPORATED PER SQ. FT. OF B.H.S. PER HOUR

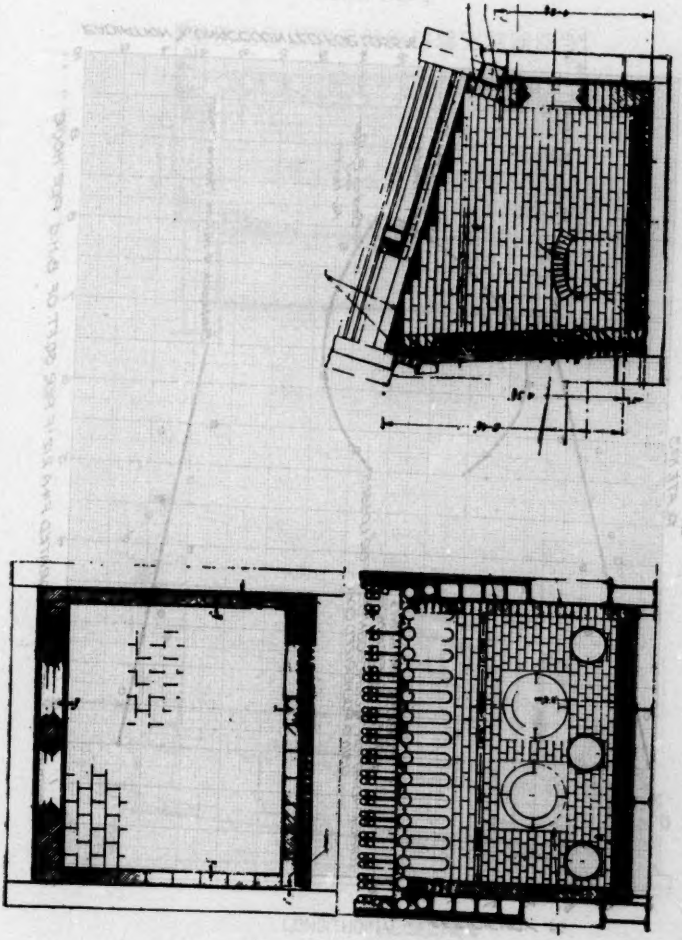


PLATE No. 2.

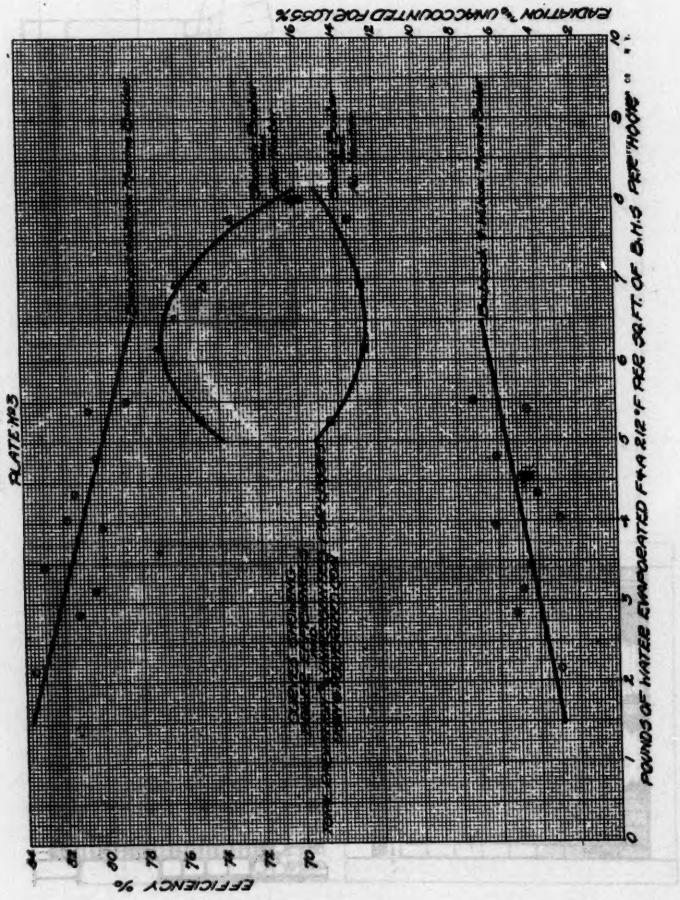
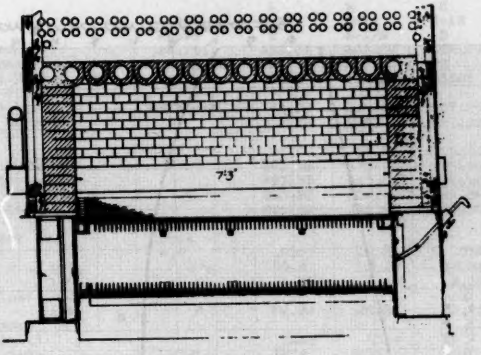
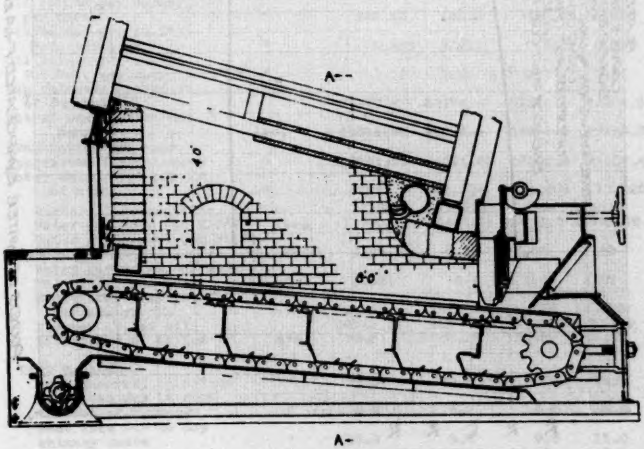

RADIATION % UNACCOUNTED FOR LOSS %
POUNDS OF WATER EVAPORATED PER 212° F PER SQ. FT. OF B.H.S. PER HOUR
EFFICIENCY %
PLATE N°3

PLATE No. 4.



FRONT VIEW SECTION A-A



A-
LONGITUDINAL SECTION

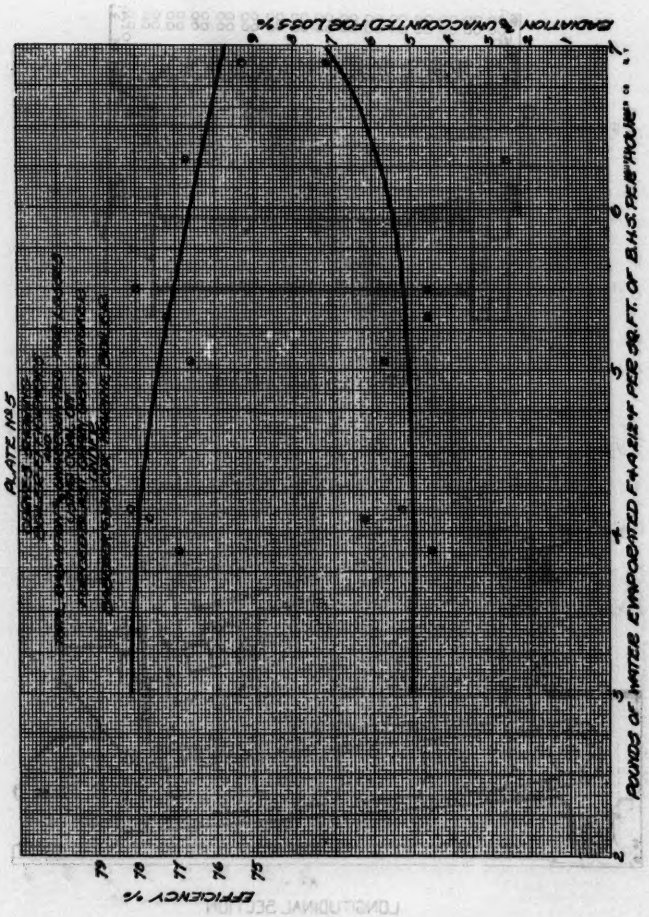

RADIATION & UNACCOUNTED FOR LOSS %
POUNDS OF WATER EVAPORATED PER SQ. FT. OF B.H.S. PER HOUR
EFFICIENCY %
PLATE No. 5

TABLE No. 1

Evaporation Tests of a Scotch Boiler fitted with
Babcock & Wilcox 15° Mayflower Oil Burners
League Island Navy Yard, Philadelphia, Pa.
Tests made by Commander H.H. Norton, U.S.N. and
Mr. C.J. Jefferson, U.S.N.E.
Boiler Heating Surface 2555 sq.ft. Total Furnace
Volume 482 cu.ft.

Test Number			1	2	3	4
Date	1904		3-14	3-14	3-15	3-15
Duration	Hours		2	2-1/2	3	2
Furnace Volume	cu.ft.		482.89	482.89	482.89	482.89
Number of burners used			3	3	3	3
Sprayer Plate Number			8250	8220	8080	8080
PRESSURES -						
Steam in boiler	lbs/sq.in.		145.80	145.84	147.10	140.45
Oil at the burners	"		219.88	206.89	241.80	207.86
TEMPERATURES -						
Room	°F		64.0	68.37	70.08	77.05
Oil at the burners	"		160.58	176.15	188.40	173.65
Gases in the uptake	"		499.87	653.84	661.54	638.75
Feed Water	"		280.88	305.42	301.21	305.70
Steam at calorimeter	"		296.64	296.46	298.00	298.10
DRIFTS -						
Base of stack	inches of H_2O		.056	.096	.094	.071
Gas Analysis -	CO_2	c/o	10.2	12.25	13.86	13.21
	O_2	"	6.91	4.49	2.18	2.99
Uptake	CO	"	0	0	0	0.03
	N_2	"	82.89	83.26	83.96	83.77
Quality of Smoke (Ringelman scale)			None	None	0	0
Total oil burned		lbs.	1478	2867	4025	2349
Total water fed to boiler		"	21730.5	39190.0	55765.0	32247.5
CALCULATED RESULTS -						
Quality of steam leaving boiler		c/o	99.46	99.48	99.58	99.43
Oil burned per hour		lbs.	739.0	1066.8	1341.66	1634.50
Oil burned per burner per hour.		"	246.33	355.6	447.22	541.50
Oil burned per sq.ft. H.S. per hour.		"	0.289	0.417	0.525	0.636
Oil burned per cu.ft. Fur.Vol. per hour.		"	1.53	2.20	2.77	3.36
Btu released per cu.ft. of Fur.Vol.			32370.8	40794.6	51364.1	62304.5
Actual evaporation per hour		lbs.	10860.80	15676.0	19686.0	22623.75
Evaporation per hour (corrected for moisture)		"	10801.55	15594.48	19606.8	22517.4
Water evaporated per lb. of oil.		"	14.626	14.604	14.604	13.820
Factor of evaporation			1.0559	1.0522	1.0546	1.0523
Water per hour F&A 212°F		lbs.	11200.13	16400.51	20610.9	23696.08
Water evaporated per sq.ft. H.S. per hour		"	4.38	6.13	7.66	8.88
Water per sq.ft. of H.S. F&A 212°F. per hour		"	4.36	6.4	8.06	9.27
Equivalent evaporation per lb. of oil		"	15.186	15.381	15.362	14.806
Btu per lb. of oil			18543.0	18543.0	18543.0	18543.0
Efficiency of boiler		c/o	79.31	80.49	80.39	76.38
HEAT BALANCE -						
Heat absorbed by boiler		"	79.31	80.5	80.4	76.3
Heat loss due to combustion of Hydrogen		"	6.2	6.2	6.2	6.4
Heat loss due to dry chimney gases		"	10.9	9.8	9.6	11.0
Heat loss due to CO		"	0	0	0	0.1
Radiation and unaccounted for losses		"	3.6	3.5	3.8	6.2
Total		"	100.0	100.0	100.0	100.0
OIL ANALYSIS -						
Moisture		"	0	0	0	0
Hydrogen		"	10.97	10.97	10.97	10.97
Carbon		"	86.47	86.47	86.47	86.47
Sulphur		"	2.15	2.15	2.15	2.15
Gravity-Baume'		Degrees	17.25	17.25	17.25	17.25
Heat value per lb.		B.t.u.	18543	18543	18543	18543

size of the coal fed to the stoker. At no time was difficulty experienced because of low fusing ash in the coal.

In closing it is gratifying to note the increased interest operators of modern steamships are taking in the movement to obtain more efficient combustion conditions in their furnaces. Present day competition has developed the realization that it is not only necessary to employ higher pressure and higher temperature steam, but that it is equally important that modern combustion equipment be used in generating this steam, to insure the most economical use of the fuel when it is fired in the small furnaces available under marine boilers.

TABLE No. 8.

Progressive Tests of a Babcock & Wilcox Express Type Boiler,
Fitted with 11 Babcock & Wilcox Cyclone Type Oil Burners, - Longue
Toland Navy Yard - Philadelphia, Pa. Tests made by Commander A.H. Pyne, U.S.N.
Boiler Heating Surface-7665 sq.ft. Superheating Surface-755 sq.ft.
Furnace Volume-761 cu.ft.

Test Number		30-14 C-9	30-15 C-9	30-16 C-10	30-17 C-10	30-18 C-11	30-19 C-11	30-20 C-11
1912								
Duration	hours	3	3	3	3	3	2½	2
Number of burners used		4	4	11	11	4	9	11
Steam pipe size		104	104	104	104	103	103	103
Pressures								
Steam	lbs/sq.in.	296.1	295.8	295.8	295.8	297.1	296.0	295.8
Calorimeter	"	294.1	292.2	293.2	291.6	291.6	---	---
After boiler stop	"	290.1	288.2	284.2	282.6	284.9	278.3	272.9
Air burners	"	197.2	197.6	197.3	246.9	168.4	170.6	168.3
Draft in fireroom	inches of H_2O	4.027	5.027	8.782	8.548	2.726	4.062	5.827
Draft in furnace	"	.970	1.011	5.306	5.707	.116	1.188	1.881
Draft in uptake	"	.003	.044	.192	.242	.115	.038	.046

	lbs/sq.in.	296.1	296.8	298.2	296.8	297.2	296.0	295.8
	"	284.1	282.2	283.2	281.6	281.6	---	---
r boiler slug	"	280.1	280.8	284.2	285.8	284.9	278.8	272.8
burners	"	197.2	197.6	197.3	245.9	169.4	170.6	169.2	
r in fireroom	inches of H_2O	4.087	5.867	8.782	8.548	2.728	4.962	5.827
in furnace	"	.370	1.811	1.388	5.707	.116	1.188	1.881
in uptake	"	-.005	-.048	+.182	+.242	-.115	-.038	+.044
TURES								
	of	101.7	99.8	97.8	100.1	118.1	118.8	114.6
burners	"	119.4	119.8	120.8	120.0	119.8	120.0	120.0
	"	652.0	614.0	718.0	722.0	490.0	617.0	648.0
ter	"	197.6	196.9	196.8	196.8	196.1	194.7	197.2
meter	"	315.8	314.7	302.4	296.7	315.5	---	---
leaving superheater	"	488.2	490.6	486.6	484.8	480.8	484.9	483.6
YSIS								
CO_2	%	13.82	12.87	12.76	13.01	12.90	13.42	12.44
O_2	"	5.37	4.70	5.71	5.12	5.53	4.78	6.40
CO	"	.11	.02	.00	.09	.04	.02	.00
N	"	80.80	83.31	83.53	83.86	83.53	83.78	83.16
of smoke-Ringelmann scale		0-1	0-1	1-2	1½-2	0-1	0-1	0-1
il burned	lbs.	111798.0	147782.0	206893.0	227710.0	88217.0	119775.0	115620.0
ter evaporated	"	1189960.0	1588804.0	2118254.0	2264934.0	1276640.0	1808820.0	1821384.0
ED RESULTS								
ity of steam at S.H. inlet	%	99.28	99.26	98.66	98.29	99.26	---	---
rned per hour	lbs.	3726.0	4984.0	10251.0	11585.0	2830.0	6390.0	7810.0
rned per burner per hour.	"	931.8	930.7	930.1	1032.9	709.8	710.0	710.0
rned per sq.ft. of H.S. per hour	"	.493	.736	1.362	1.601	.378	.848	1.032
rned per cu.ft. of Fur.Vol. per hour	"	4.961	7.428	13.625	15.129	3.780	8.500	10.389
released per cu.ft of Fur.Vol. per hour	"	98767.3	143498.8	262923.9	291816.0	72984	164223.7	200700.7
evaporated per hour	"	85323.0	79832.0	139452.0	152407.0	42523.0	92208.0	110643.0
evaporated per sq.ft. of H.S. per hour	"	7.313	10.526	18.434	20.256	5.621	12.189	14.628
evaporated per lb. of oil	"	14.848	14.241	13.630	13.818	14.978	14.430	14.169
ated sat. steam temp.	of	419.0	419.7	407.8	403.7	420.2	415.8	413.8
ee superheat at S.H. outlet	"	46.3	62.9	78.7	80.8	39.5	67.4	89.8
r of evaporation		1.1086	1.1161	1.1222	1.1206	1.1041	1.1204	1.1197
r per hour FRANKF'T.	lbs.	61165.1	88877.3	154493.0	171975.1	46949.4	103309.8	123798.7
r per sq.ft. of H.S. FRANKF'T per hour	"	6.026	11.748	20.687	22.732	6.206	13.657	16.364
valent evaporation per lb. of oil	"	16.456	15.917	15.296	15.148	16.537	16.167	15.851
ght of dry gases per lb. of oil	"	16.49	17.86	16.94	16.49	16.71	17.34	17.36
ght of dry air per lb. of oil	"	15.89	17.24	17.04	16.58	16.90	17.41	17.40
per lb. of oil		19300	19300	19300	19300	19280	19280	19280
er efficiency	%	82.54	80.03	76.91	76.15	83.24	81.37	79.75
ALANCE								
absorbed by boiler & superheater	%	82.54	80.03	76.91	76.15	83.24	81.37	79.98
loss due to combustion of hydrogen	"	6.72	8.84	7.15	7.12	6.37	6.89	6.77
loss due to dry chimney gases	"	8.82	11.43	13.00	12.71	7.80	10.83	11.85
loss due to CO	"	.36	.07	.00	.31	.14	.11	.00
ation and unaccounted for losses	"	1.54	1.63	2.94	3.71	2.45	1.00	1.22
		100.00	100.00	100.00	100.00	100.00	100.00	100.00
sis of the Oil								
rbon	%	86.90	86.90	86.90	86.90	86.90	86.90	86.90
drogen	"	11.70	11.70	11.40	11.40	11.40	11.40	11.40
lphur	"	.86	.86	.86	.86	.86	.86	.86
Nitrogen	"	.24	.24	.24	.24	.29	.29	.29
Oxygen and undetermined	"	1.41	1.41	1.41	1.41	1.45	1.45	1.45
Gravity - Baume	Degrees	26.1	26.1	26.56	26.56	26.56	26.56	26.56

TABLE No. 2

Evaporation Tests of a Babcock & Wilcox Marine Boiler
Fitted with 3 Babcock & Wilcox 12' Mayflower Type Oil Burners
Tests made by Mr. T.B. Stillman and R.L. Boland - B.&W. Co.
Boiler Heating Surface 2765 sq.ft. Superheating Surface 564 sq.ft.
Furnace Volume - 250 cu.ft.

Test No.			A 1h	Adank	A V
Date		1926	6-27	6-8	7-30
Duration		Hours	7	2	6-1/2
Type of burner			←——12"-Mayflower——→		
PRESSURES					
Steam-					
Drum	lbs/sq.in.		207.0	205.0	215.4
Calorimeter	"		200.0	200.0	212.4
Superheater					
Outlet	"		201.0	206.0	206.7
Burner turbines	"		---	180.0	198.7
Oil at burners	"		189.0	216.1	174.9
TEMPERATURES					
Room	°F		135.2	81.0	108.7
Oil at burners	"		200.1	200.2	206.1
Uptake gas	"		434.0	427.4	527.7
Feed Water	"		70.2	70.0	72.0
Steam at calori-meter	"		308.7	304.0	307.7
Steam at super-heater outlet	"		456.3	461.8	468.7
DRAFTS					
Room pressure	in. of H_2O		---	---	0.100
Furnace	" " "		.207	.021	1.575
Uptake	" " "		.804	.998	4.547
GAS ANALYSIS					
Top of tube	CO_2	%	12.02	14.02	13.62
	O_2	"	5.72	2.52	1.62
	CO	"	0	0	0
	N_2	"	82.45	83.46	84.76
Uptake	CO_2	"	12.52	13.02	12.52
	O_2	"	5.24	3.08	4.66
	CO	"	0	0	0
	N_2	"	82.20	83.27	82.84
Quality of smoke			Ringelmann scale	0-1/4	0-1/4	0-2/4
Total oil burned		lbs.	5614.7	2008.4	2020.0
Total water evaporated		"	49477.0	18000.0	16027L.0
CALCULATED RESULTS					
Quality of steam at B.H. inlet		%	99.70	99.02	98.79
Total water fed to boiler		lbs.	50,000.	18,000.	16,000.
Total coal as fired		"	5000.0	1800.0	1600.0
Total refuse		"	500.0	180.0	160.0
Moisture in the coal as fired		%	2.0	2.0	2.0
Coal burned per hour as fired		lbs.	714.3	900.0	800.0
Coal per burner per hour - as fired		"	238.1	300.0	266.7
Coal per hour-dry		"	700.0	882.0	784.0
Coal per burner per hour-dry		"	233.3	294.0	261.3
Coal per sq.ft. of H.S. per hour as fired		"	.086	.108	.096
Coal per sq.ft of H.S. per hour-dry		"	.084	.106	.094
Coal per sq.ft. of Fur. Vol. per hour as fired		"	2.86	3.60	3.20
Coal per sq.ft. of Fur. Vol. per hour-dry		"	2.80	3.53	3.14
Btu released per sq.ft. of Fur.Vol. per hour			32100.0	40607.0	36080.0
Actual evaporation per hour		"	10000.0	12677.0	10021.0
Quality of steam leaving boiler		%	99.70	99.02	98.79

TABLE No. 4

Pulverized Coal Evaporation Tests of a Scotch Boiler
Fitted with Peabody Burners and using Kennedy-Van Saun Mill -
League Island Navy Yard - Philadelphia, Pennsylvania
Tests made by Commander J.S. Brown, U.S.N. and Mr. C.J. Jefferson
U.S.N.R.
Boiler Heating Surface 6666 sq.ft. Total Furnace Volume 470 cu.ft.
Air Heating Surface 636 sq.ft.

Test No.			1	2	3	4	5
Date started		1927	7-27	7-28	7-31	8-2	8-4
Date terminated		1927	7-28	7-31	8-2	8-4	8-5
Duration		Hours	48	48	48	48	48
PRESSURES							
Steam in boiler (absolute)		lbs/sq.in.	190.3	190.0	190.0	191.8	192.1
Air - Primary							
Discharge from fan		in. of H_2O	1.69	1.63	1.97	2.19	2.09
Burner Tuyere		" " "	.96	.89	1.11	1.26	1.15
Air - Secondary		" " "					
In double front		" " "	.55	.43	.56	1.05	1.03
TEMPERATURES							
Room		°F	101.0	96.0	98.0	91.0	95.0
Primary air -							
At feeder end of mill		"	148.0	134.0	138.0	153.0	155.0
Inlet to fan		"	150.0	142.0	142.0	147.0	148.0
At distributor		"	144.0	147.0	140.0	140.0	136.0
Secondary air -							
Entrance to air heater		"	89.0	88.0	86.0	88.0	93.0
Air ducts leaving air heater		"	189.0	188.0	188.0	201.0	198.0
At burners		"	214.0	217.0	218.0	211.0	208.0
Gas leaving boiler		"	533	531	532	535	532
Leaving air heater		"	338	334	332	338	338
DRAFTS							
Above air heater		in. of H_2O	.28	.248	.248	.286	.248
Below air heater		" " "	.28	.248	.237	.284	.238
Rear of furnace		" " "	.178	.097	.068	.099	.100
GAS ANALYSIS							
Uptake	CO_2	%	16.18	16.72	17.02	17.18	16.98
	O_2	"	2.68	1.97	1.72	1.97	2.78
	CO	"	.000	0	.027	.079	.013
	N_2	"	81.17	81.28	81.23	81.88	80.99
CALCULATED RESULTS							
Total water fed to boiler		lbs.	618,816.	710,272.	611,296.	992,992	988,768
Total coal as fired		"	37800.0	44188.0	38797.0	61888.0	61888.0
Total refuse		"	3020.0	4068.0	4376.0	4881.0	5708.0
Moisture in the coal as fired		%	2.3	2.0	2.0	2.4	2.0
Coal burned per hour as fired		lbs.	1666.4	1668.3	1600.4	1660.0	1672.2
Coal per burner per hour - as fired		"	421.6	421.0	423.2	420.8	420.8
Coal per hour-dry		"	1100.1	1264.2	1568.2	1707.0	1618.0
Coal per burner per hour-dry		"	268.4	321.4	319.4	326.7	329.0
Coal per sq.ft. of H.S. per hour as fired		"	.472	.241	.238	.781	.770
Coal per sq.ft of H.S. per hour-dry		"	.421	.220	.233	.238	.781
Coal per cu.ft. of Fur. Vol. per hour as fired		"	2.87	2.91	2.40	2.82	4.20
Coal per cu.ft. of Fur. Vol. per hour-dry		"	2.21	2.20	2.26	2.24	4.00
Btu released per cu.ft. of Fur.Vol. per hour			27100.0	42827.0	40978.0	42843.0	42824.0
Actual evaporation per hour		"	12880.0	14477.0	12681.0	12888.0	12488.0
Quality of steam leaving boiler		%	99.80	99.81	99.84	99.88	99.84

Singleness scale		0-1/4	0-1/4	0-1/4	
Total oil burned	lbs.	5214.7	9282.4	9281.8	
Total water evaporated	"	69477.0	126888.0	126874.0	
BOILERROOM READINGS					
Quality of steam at S.H. inlet	o/o	99.78	99.68		99.74
Oil burned per hour	lbs.	716.4	1248.7		1868.9
Oil burned per burner per hour	"	238.8	417.9		619.6
Oil burned per sq.ft.H.S. per hour	"	.860	.488		.678
Oil burned per cu.ft. F.V. per hour	"	2.115	8.648		8.682
Lbs released per cu.ft. of Fur. Vol. per hour	"	27440.9	197827.8		1240819.8
Water evaporated per hour	"	9682.4	16988.9		28740.8
Water evaporated per lb. of oil	"	18.097	18.596		18.528
Calculated sat. steam temp.	deg	209.78	209.18		209.44
Degrees superheat at S.H. outlet	"	48.22	71.28		88.24
Factor of evaporation		1.2187	1.2269		1.2202
Total water evap. Rankine F	lbs.	88841.8	41876.1		127983.8
Water per hour Rankine F	"	11424.4	20128.1		29440.9
Water per sq.ft. HS Rankine F per hour	"	4.180	7.690		10.888
Equivalent evaporation per lb. of oil	"	15.861	15.888		15.899
Btu per lb. of oil		18443	18480		18504
Boiler efficiency	o/o	83.66	81.92		79.72
HEAT BALANCE					
Heat absorbed by boiler & superheater	o/o	83.66	81.92		79.71
Heat loss due to combustion of hydrogen	"	6.88	6.86		7.07
Heat loss due to dry chimney gases	"	6.49	8.40		9.78
Radiation & unaccounted for losses	"	3.31	3.18		3.18
Total		100.00	100.00		100.00
ULTIMATE ANALYSIS of the OIL					
Carbon	o/o	84.92	84.92		85.82
Hydrogen	"	10.96	10.96		11.80
Sulphur	"	1.72	1.72		1.40
Oxygen & Nitrogen	"	2.40	2.40		1.38

Vol. per hour-deg.	"	2.81	2.80	2.86	2.84
Btu released per cu.ft. of Fur.Vol. per hour		37180.0	48897.0	48878.0	48861.0
Actual evaporation per hour	"	18880.0	14977.0	14981.0	14988.0
Quality of steam leaving boiler	o/o	98.80	98.88	98.88	98.88
Evaporation per hour (corrected for moisture	lbs.	12794.0	14848.0	14889.0	14847.0
Water per hour per sq.ft. of H.S.	"	8.01	8.86	8.81	7.88
Water per lb. of fuel-as fired	"	10.61	10.81	10.84	10.68
Factor of evaporation		1.1888	1.1888	1.1888	1.1888
Water per hour Rankine F.	"	12881.0	12780.0	12780.0	12880.0
Water per sq.ft. H.S. Rankine F per hour	"	8.06	8.16	8.08	7.97
Equivalent evaporation per lb. of coal-as fired	"	11.16	11.86	11.87	11.88
Equivalent evaporation per lb. of coal-dry	"	11.61	11.61	11.87	11.86
Btu per lb. of coal-as fired		14660	14580	14578	14588
Efficiency of boiler and grate	o/o	76.00	77.06	76.78	76.86
HEAT BALANCE					
Heat absorbed by boiler &	"	76.00	77.06	76.78	76.86
Heat loss due to moisture in coal	"	.17	.84	.82	.82
Heat loss due to com-bustion of hydrogen	"	8.68	8.64	8.70	8.72
Heat loss due to dry chimney gases	"	6.66	7.68	9.65	9.10
Heat loss due to CO	"	.08	.0	.18	.08
Radiation & unaccounted	"	14.88	12.88	12.87	12.88
TOTAL	"	100.00	100.00	100.00	100.00
COAL ANALYSIS - PROXIMATE					
Moisture	%	4.1	6.0	8.6	8.4
Fixed carbon	"	72.1	70.9	71.7	72.8
Volatile matter	"	20.1	20.1	20.0	20.4
Ash	"	8.7	6.1	8.7	8.8
COAL ANALYSIS - ULTIMATE					
Hydrogen	"	4.6	4.6	4.6	4.6
Carbon	"	81.8	80.7	81.1	82.2
Nitrogen	"	1.4	1.8	1.8	1.8
Oxygen	"	8.8	8.8	8.8	8.8
Sulphur	"	1.8	1.8	1.8	1.7
Ash	"	8.7	6.1	8.7	8.8
SCREENING.					
Thru 80 mesh	"	96.6	99.7	99.8	99.2
Thru 10 mesh	"	95.8	98.1	94.8	97.0
Thru 100 mesh	"	91.2	94.6	92.6	93.1
Thru 140 mesh	"	88.4	88.8	87.2	88.2
Thru 200 mesh	"	77.4	80.9	78.1	79.8
Thru 300 mesh	"	68.0	70.4	68.8	69.8
ASH AND REFUSE DATA					
Fusion temperature of ash	deg F	<----------2170--------->			
Combustible in ash and fine dust	o/o	<----------35(approx)--------->			
Non-combustible in ash and fine dust	"	<----------65(approx)--------->			
Heat value of ash and fine dust	B.T.U.	<----------2100--------->			

TABLE 5

PULVERIZED COAL EVAPORATION TESTS OF A BABCOCK & WILCOX MARINE TYPE BOILER FITTED WITH BABCOCK & WILCOX LODI PULVERIZED COAL BURNERS AND USING A FULLER - BONNOT BALL MILL.

RATING - No. 5.

TESTS MADE BY T. B. STILLMAN AND E. L. BOLAND OF THE BABCOCK & WILCOX CO.

BOILER HEATING SURFACE 8756 SQ. FT. FURNACE VOLUME 250 CU. FT.

1	Date of test, 1927	7/25	7/26	7/27	7/28	7/29	8/4	8/5	8/11	8/16	8/17
2	Duration (hours)	4½	7	5	6	7	7	6	5	6	6
3	Type of coal used	Dundon gas slack—West Virginia								Pocahontas	
4	Speed of mill (R. P. M.)	32	32	32	32	32	32	32	32	32	32
5	Speed of exhauster (R. P. M.)	1,670.3	1,819.3	2,258.1	1,878.4	2,247.0	2,031.0	1,842.0	2,350.0	2,336.8	2,336.0
	Steam Pressure (lbs.—sq. in.):										
6	Drum	205.1	192.3	191.0	187.1	191.0	185.1	205.3	176.3	183.3	187.0
7	Superheater inlet	203.3	189.5	188.5	186.2	189.3	184.5	205.0	176.0	182.2	186.4
8	Superheater outlet	202.0	187.0	187.3	185.6	187.6	183.9	204.7	180.2	180.7	184.7
	Temperatures (° F.):										
9	Furnace	2,126.1	2,237.3	2,302.2	2,252.1	2,291.1	2,255.9	2,076.0	2,483.5	2,450.6	2,535.3
10	Gas-uptake	420.8	434.2	445.4	442.7	479.6	440.2	424.2	471.3	448.1	454.4
11	Feed water	170.6	196.0	197.9	80.9	80.4	197.6	201.7	189.5	195.5	196.4
12	Superheated steam	438.3	431.0	455.6	444.5	458.6	428.1	429.9	442.7	451.7	449.8
13	Room	92.5	101.3	102.0	103.0	102.8	93.5	99.2	98.3	89.5	94.6
14	Air entering mill	166.5	169.0	183.8	185.9	188.5	183.9	187.2	167.6	166.5	
15	Calorimeter—superheater inlet	312.6	310.5	310.1	310.1	309.9	308.6	313.0	307.3	308.5	306.3
	Drafts (in H_2O):										
16	Furnace	.069	.086	.344	.053	.283	.049	.019	.167	.339	.568
17	Uptake	.269	.324	1.321	.538	1.299	.213	.076	.649	.873	1.206
18	Air pressure in burner pipe	2.30	2.39	4.36	2.99	4.27	2.01	1.46	2.61	2.435	2.35
	Gas Analysis (%):										
19	Top of 1st pass CO_2	13.68	13.72	13.17	13.54	13.70	13.34	13.34	13.09	13.43	13.82
20	Top of 1st pass O_2	5.53	5.29	5.89	5.51	5.22	5.66	5.66	5.89	5.55	5.28
21	Top of 1st pass CO	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
22	Top of 1st pass N_2	80.84	80.99	80.94	80.95	81.08	81.10	81.00	81.02	81.00	80.90
23	Uptake CO_2	12.57	13.22	12.40	12.35	12.39	11.25	12.68	12.00	12.16	12.57
24	Uptake O_2	6.48	6.17	6.52	6.53	6.68	7.65	6.30	6.94	7.30	6.67
25	Uptake CO	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
26	Uptake N_2	80.95	80.61	81.08	81.12	80.93	81.10	81.02	81.06	80.54	80.76
27	Quality of smoke	Light grey ash									
28	Total water evaporated (lbs.)	51,445.0	60,812.0	68,531.0	53,108.0	84,156.0	55,690.0	52,885.0	50,169.0	64,465.0	78,830.0
29	Total wet coal to burners (lbs.)	5,577.0	6,683.0	7,189.0	6,230.0	9,865.0	5,744.0	5,365.0	5,382.0	6,880.0	7,950.0
30	Moisture in coal entering mill (%)	3.55	2.48	2.84	4.82	4.27	5.06	6.22	3.53	4.55	3.83
31	Moisture in coal leaving mill (%)	1.07	.98	1.08	1.23	1.35	1.27	1.47	1.23	.63	.76
32	Quality of steam S. H. inlet (%)	99.93	99.91	99.90	99.92	99.86	99.87	99.93	99.84	99.81	99.86
33	Calculated superheat steam temperature at S. H. outlet (° F.)	388.68	382.58	382.70	382.82	382.82	381.24	389.66	379.78	379.96	381.60
34	Degrees superheat at S.H. outlet (°F.)	49.42	46.02	73.00	62.35	75.98	46.86	40.24	62.95	72.04	68.10
35	Factor of evaporation	1.1263	1.0973	1.1099	1.2243	1.2326	1.0946	1.0868	1.1119	1.1120	1.1192
36	Total water F & A 212° F. (lbs.)	55,399.8	66,729.0	76,062.6	65,014.8	103,730.7	60,939.4	55,805.2	55,772.9	71,749.5	79,272.9
37	Water evaporated per hour, actual (lbs.)	6,985.1	8,687.4	13,706.2	8,851.3	12,022.3	7,927.1	5,480.8	10,033.8	10,744.2	11,886.0
38	Water per hour F & A 212° F. (lbs.)	7,866.6	9,532.7	15,212.5	10,835.8	14,818.7	8,677.1	5,963.6	11,154.6	11,958.2	13,212.1

33	Calculated extreme of steam temperature at S. H. outlet (° F.)	388.48	382.58	382.70	382.02	382.82	381.24	389.66	379.78	379.96	381.60
34	Degrees superheat at S.H. outlet (° F.)	49.42	46.02	75.00	62.38	75.98	46.86	60.24	62.92	75.94	68.10
35	Factor of evaporation	1.1263	1.0975	1.1099	1.1243	1.1526	1.0946	1.0885	1.1119	1.1130	1.1192
36	Total water F & A 212° F. (lbs.)	35,399.6	66,729.0	76,052.6	65,014.8	103,730.7	60,730.4	33,803.2	55,712.9	71,349.3	79,273.9
37	Water evaporated per hour, actual (lbs.)	6,985.1	8,667.4	13,708.2	8,851.3	13,022.3	7,927.1	5,480.8	10,032.9	10,744.2	11,885.0
38	Water per hour F & A 212° F. (lbs.)	7,844.6	9,532.7	13,212.5	10,535.8	14,818.7	8,677.1	5,067.5	11,154.5	11,958.5	13,212.2

39	Water per hour F & A 212° F. per sq. ft. of H. S. (lbs.)	2.86	3.46	3.52	3.93	5.38	3.15	2.17	4.05	4.341	4.796
40	Total dry coal (lbs.)	3,257.17	5,912.6	6,984.8	5,929.7	9,443.8	5,453.4	3,155.7	5,076.3	6,042.0	6,773.0
41	Wet coal per hour (lbs.)	750.4	866.1	1,437.8	1,038.3	1,409.3	820.6	560.8	1,052.4	1,055.0	1,175.0
42	Dry coal per hour (lbs.)	723.8	844.7	1,397.0	988.3	1,349.1	779.1	526.0	1,015.3	1,007.0	1,128.8
43	Dry coal burned per sq. ft. of F. V. per hour (lbs.)	2.90	3.38	5.59	3.95	5.40	3.12	2.10	4.06	4.03	4.52
44	Dry coal burned per sq. ft. of H. S. per hour (lbs.)	.263	.307	.507	.359	.490	.283	.191	.369	.366	.410
45	Wet coal per burner per hour (lbs.)	375.2	433.05	718.9	519.2	704.7	410.3	280.4	526.2	527.5	587.5
46	Dry coal per burner per hour (lbs.)	361.9	422.4	698.5	494.2	674.6	389.6	263.0	507.7	503.5	564.4
47	B. T. U. liberated per cu. ft. of F. V. per hour	37,643.0	44,033.9	73,062.5	52,507.4	71,306.4	41,939.0	27,680.1	53,997.7	57,036.6	63,876.6
48	Actual evaporation per lb. of dry coal (lbs.)	9.65	10.28	9.81	8.96	8.91	10.17	10.42	9.88	10.67	10.46
49	Equivalent evaporation per lb. of dry coal (lbs.)	10.87	11.29	10.89	10.96	10.98	11.16	11.35	10.99	11.88	11.70
50	Total air supplied per hour (lbs.)	9,880.6	11,561.4	20,199.2	13,826.6	18,889.6	1,099.8	7,353.5	14,224.4	15,014.4	16,589.5
51	Primary air per hour (lbs.)	.....	7,372.9	8,825.4	7,117.3	7,998.2	7,000.8	4,327.2	.....	7,812.0	7,574.4
52	Per cent primary air	.....	63.77	43.69	51.35	41.33	63.14	58.85	.....	50.70	45.74
53	Per cent excess air used for combustion	36.98	32.88	38.05	36.74	32.24	35.94	36.01	37.88	35.06	32.84
54	Horsepower input, mill	20.7	20.8	24.2	21.0	23.6	20.8	20.8	20.9	20.8	21.0
55	Horsepower input, exhauster	5.8	4.5	18.7	6.1	17.2	4.2	5.2	6.2	7.1	7.3
56	Total H. P. of mill and exhauster	26.5	25.3	42.9	27.1	40.8	25.0	26.0	27.1	27.9	28.3
57	Total H. P. of mill and exhauster per 1,000 lb. wet coal per hour	32.64	29.29	29.83	26.10	28.91	30.47	42.79	25.75	26.48	24.09
58	B. T. U. per lb. of dry coal	12,980	12,205	13,428	13,393	13,216	13,442	13,181	13,029	14,133	14,132
59	Efficiency—boiler, superheater and furnace (%)	81.27	82.97	78.70	80.01	80.62	80.42	83.56	81.85	81.46	80.34
	Heat Balance (%):										
60	Heat absorbed by boiler	81.27	82.97	78.70	80.01	80.62	80.42	83.56	81.85	81.46	80.34
61	Loss due to moisture in coal	0.34	0.23	0.26	0.45	0.41	0.68	0.00	0.34	0.41	0.35
62	Loss due to water from combustion of H_2	4.11	4.01	4.01	4.01	4.09	3.99	4.03	4.15	3.27	3.24
63	Loss due to dry chimney gases	9.10	8.40	9.87	9.35	10.41	10.31	8.31	10.77	10.85	10.01
64	Radiation and unaccounted for losses	5.16	4.39	7.16	6.18	4.47	4.80	3.80	2.89	4.01	6.06
65	Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
	Pleasant Coal (%):										
66	Through 48 mesh	99.8	99.8	99.6	99.6	99.6	.....	99.8	99.8	99.8	99.8
67	Through 100 mesh	97.4	94.8	93.4	93.0	89.4	.....	94.2	93.4	97.0	96.0
68	Through 200 mesh	88.0	78.0	75.6	72.0	66.2	.....	89.0	74.6	72.6	77.6
	Furnace Ash Analysis (%):										
69	Combustible	.....	0.13	0.05	0.80	0.00	.....	0.36	0.50	0.07	0.00
70	Ash	.....	99.85	99.95	99.20	100.00	.....	99.64	99.50	99.93	100.00
	Uptake Dust Analysis (%):										
71	Combustible	7.47	6.39	15.16	9.46	23.84	9.53	4.43	6.23	16.22	14.88
72	Ash	92.53	93.61	84.83	90.53	76.16	90.87	95.58	91.77	83.78	85.12

COAL ANALYSIS.

ULTIMATE ANALYSIS OF DRY COAL.

Dundee gas slack

Pocahontas screened

19	Top of 1st pass	CO_2	13.63	13.72	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
20		O_2	5.53	5.29	5.89	5.51	5.22	5.66	5.66	5.89	5.53
21		CO	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
22		N_2	80.84	80.99	80.94	80.95	81.08	81.10	81.00	81.02	81.00	80.90
23	Uptake	CO_2	12.57	13.22	12.40	12.33	12.39	11.25	12.68	12.00	12.10	12.57
24		O_2	6.48	6.17	6.52	6.53	6.68	7.65	6.30	6.94	7.30	6.67
25		CO	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
26		N_2	80.95	80.61	81.08	81.12	80.93	81.10	81.02	81.06	80.54	80.76
27	Quality of smoke						Light gray ash					
28	Total water evaporated (lbs.)		31,443.0	60,812.0	66,531.0	53,108.0	84,156.0	55,490.0	32,885.0	50,160.0	64,465.0	70,830.0
29	Total wet coal to burners (lbs.)		8,377.0	6,063.0	7,189.0	6,230.0	9,865.0	5,744.0	3,365.0	5,282.0	6,380.0	7,080.0
30	Moisture in coal entering mill (%)		3.55	2.48	2.84	4.82	4.27	5.06	6.22	3.53	4.55	3.93
31	Moisture in coal leaving mill (%)		1.07	.94	1.08	1.22	1.35	1.27	1.47	1.23	.63	.76
32	Quality of steam S. H. inlet (%)		99.93	99.91	99.90	99.92	99.88	99.87	99.93	99.84	99.81	99.86
33	Calculated saturated steam temperature at S. H. outlet (° F.)		388.68	382.58	382.70	382.02	382.82	381.24	389.66	379.75	379.96	381.60
34	Degrees superheat at S.H.outlet(°F.)		49.42	48.02	73.00	62.38	75.98	46.86	40.24	62.95	75.94	69.10
35	Factor of evaporation		1.1262	1.0973	1.1099	1.2242	1.2326	1.0946	1.0888	1.1119	1.1130	1.1192
36	Total water F & A 212° F. (lbs.)		35,399.8	66,729.0	76,062.6	65,014.8	105,730.7	60,739.4	35,805.2	55,772.9	71,749.5	79,272.9
37	Water evaporated per hour, actual (lbs.)		6,985.1	8,687.4	13,706.2	8,851.3	12,022.3	7,927.1	5,480.8	10,032.0	10,744.2	11,805.0
38	Water per hour F & A 212° F. (lbs.)		7,866.6	9,532.7	15,212.5	10,835.8	14,818.7	8,677.1	5,967.5	11,154.5	11,958.3	13,212.2

39	Water per hour F & A 212° F. per sq. ft. of H. S. (lbs.)	2.86	3.46	5.52	3.93	5.38	3.15	2.17	4.05	4.341	4.796
40	Total dry coal (lbs.)	3,257.17	5,912.6	6,984.8	5,929.7	9,643.2	5,453.4	3,155.7	5,076.3	6,042.0	6,773.0
41	Wet coal per hour (lbs.)	750.4	866.1	1,437.8	1,038.3	1,409.3	820.6	560.8	1,052.4	1,055.0	1,175.0
42	Dry coal per hour (lbs.)	723.8	844.7	1,397.0	988.3	1,349.1	779.1	526.0	1,015.3	1,007.0	1,128.8
43	Dry coal burned per cu. ft. of F. V. per hour (lbs.)	2.90	3.38	5.59	3.95	5.40	3.12	2.10	4.06	4.03	4.52
44	Dry coal burned per sq. ft. of H. S. per hour (lbs.)	.263	.307	.507	.359	.490	.283	.191	.369	.366	.410
45	Wet coal per burner per hour (lbs.)	375.2	433.05	718.9	519.2	704.7	410.3	280.4	526.2	527.5	587.5
46	Dry coal per burner per hour (lbs.)	361.9	422.4	698.5	494.2	674.6	389.6	263.0	507.7	503.5	564.4
47	B. T. U. liberated per cu. ft. of F. V. per hour	37,642.0	44,632.9	75,062.5	52,507.4	71,366.4	41,939.0	27,680.1	52,897.7	57,036.6	63,876.6
48	Actual evaporation per lb. of dry coal (lbs.)	9.65	10.28	9.81	8.96	8.91	10.17	10.42	9.88	10.67	10.46
49	Equivalent evaporation per lb. of dry coal (lbs.)	10.87	11.29	10.89	10.96	10.98	11.16	11.35	10.99	11.88	11.70
50	Total air supplied per hour (lbs.)	9,800.6	11,561.4	20,199.2	13,806.6	18,389.6	1,099.8	7,333.5	14,224.4	15,014.4	16,359.5
51	Primary air per hour (lbs.)	.....	7,372.9	8,825.4	7,117.3	7,998.2	7,006.8	4,527.2	.....	7,612.0	7,574.4
52	Per cent primary air	.....	63.77	43.69	51.55	41.32	63.14	58.85	.....	50.70	45.74
53	Per cent excess air used for combustion	34.98	32.88	58.05	34.74	32.34	35.94	36.01	37.88	35.06	32.84
54	Horsepower input, mill	20.7	20.8	24.2	21.0	23.6	20.5	20.8	20.9	20.8	21.0
55	Horsepower input, exhauster	3.8	4.5	18.7	6.1	17.2	4.2	3.2	6.2	7.1	7.3
56	Total H. P. of mill and exhauster	24.5	25.3	42.9	27.1	40.8	25.0	24.0	27.1	27.9	28.3
57	Total H. P. of mill and exhauster per 1,000 lb. wet coal per hour	32.64	29.29	29.83	26.10	28.91	30.47	42.79	25.75	26.45	24.09
58	B. T. U. per lb. of dry coal	12,980	13,205	13,428	13,393	13,216	13,442	13,181	13,039	14,153	14,132
59	Efficiency—boiler, superheater and furnace (%)	81.27	82.97	78.70	80.01	80.62	80.42	83.56	81.85	81.46	80.34
	Heat Balance (%):										
60	Heat absorbed by boiler	81.27	82.97	78.70	80.01	80.62	80.42	83.56	81.85	81.46	80.34
61	Loss due to moisture in coal	0.34	0.23	0.26	0.45	0.41	0.68	0.60	0.34	0.41	0.35
62	Loss due to water from combustion of H_2	4.11	4.01	4.01	4.01	4.09	3.99	4.03	4.15	3.27	3.24
63	Loss due to dry chimney gases	9.10	8.46	9.87	9.35	10.41	10.31	8.81	10.77	10.85	10.01
64	Radiation and unaccounted for losses	5.18	4.39	7.16	6.18	4.47	4.60	3.00	2.89	4.01	6.06
65	Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
	Fineness of Coal (%):										
66	Through 48 mesh	97.8	99.8	99.6	99.6	99.6	.....	99.8	99.8	99.8	99.8
67	Through 100 mesh	97.4	94.8	93.4	93.0	89.4	.....	98.2	93.4	97.0	96.0
68	Through 200 mesh	83.0	78.0	75.6	72.0	66.2	.....	89.0	74.6	72.6	77.6
	Furnace Ash Analysis (%):										
69	Combustible	.....	0.15	0.05	0.80	0.00	.....	0.36	0.50	0.07	0.00
70	Ash	.....	99.85	99.95	99.20	100.00	.....	99.64	99.50	99.93	100.00
	Uptake Dust Analysis (%):										
71	Combustible	7.47	6.39	15.18	9.65	21.84	9.93	4.42	6.23	16.22	14.88
72	Ash	92.53	93.61	84.82	90.35	78.16	90.07	95.58	93.77	83.78	85.12

70	Ultimate Coal Analysis (%):										
71	Combustible	7.47	6.39	15.18	9.05	23.84	9.83	4.42	6.23	16.22	14.88
72	Ash	92.53	93.61	84.82	90.95	76.16	90.07	95.58	93.77	83.78	85.12

COAL ANALYSES

ULTIMATE ANALYSIS OF DRY COAL

	Dundon gas slack Dundon, W. Va.	Pocahontas screened coal West Virginia.
Carbon	74.75	81.80
Hydrogen	4.97	4.24
Nitrogen	1.30	1.18
Sulphur	.78	.88
Ash	11.79	10.80
Oxygen	6.38	2.30

PROXIMATE ANALYSIS

Moisture	4.94	4.24
Volatile matter	34.50	16.23
Fixed Carbon	53.71	73.67
Ash	11.79	10.20
Sulphur	.78	.59
Fusing point of ash F	2,710	2,705

WEIGHTS AND PERCENTAGES OF FURNACE ASH

Date	7/21-7/22	7/25-7/29	8/3-8/6	8/10-8/12	8/16-8/19
Number of hours of operation	12.00	106.33	77.33	53.75	77.00
Total wet coal fired, lbs.	14,245	95,464	62,884	50,617	74,505
Average per cent of ash in coal	14.76	11.65	12.17	13.98	9.23
Total ash in coal, lbs.	2,102.6	11,121.1	7,650.5	7,076.3	6,876.2
Total slag removed from furnace, lbs.	450	2,070	1,000	904	1,373
Average per cent of ash in slag	99.75	99.75	99.72	99.55	99.98
Total ash in slag removed from furnace, lbs.	448.9	2,064.8	1,057.8	899.9	1,372.7
Percentage of total ash fired remaining in furnace	21.34	18.55	13.82	12.72	19.96

TABLE NO. 8

EVAPORATION TESTS OF A BABCOCK & WILCOX MARINE TYPE WATERTUBE BOILER FITTED WITH BABCOCK & WILCOX FORCED BLAST CHAIN GRATE STOKER

TESTS BY E. L. BOLAND OF THE BABCOCK & WILCOX CO.

BOILER HEATING SURFACE 2706 SQ. FT. FURNACE VOLUME 167 CU. FT. GRATE SURFACE 56 SQ. FT.

1	Date of test, 1923	7/2	7/3	7/6	9/11	9/12	9/28	9/14	9/18	9/25	9/26	9/27	
2	Duration of test (hours)	6	5	6	5	6	6.5	6	6	6.5	6	6	
3	Type of coal used	Dunston Gas Black								Pocahontas			
4	Speed of stoker (inches per min.)	2.22	3.99	2.27	3.22	2.82	2.23	2.82	2.82	2.82	3.32	2.27	
5	Height of gate (inches)	4.0	4.5	4.0	4.5	4.0	4.0	4.0	4.0	4.0	4.5	4.0	
	Steam pressure (lbs.—sq. in):												
6	Drum	196.7	201.4	191.4	191.5	193.2	194.9	196.8	189.6	193.9	198.8	195.2	
7	Superheater inlet	195.7	198.8	189.3	187.9	191.0	193.7	193.0	187.8	191.7	196.2	193.6	
8	Superheater outlet	194.7	196.4	188.3	186.5	189.5	192.6	193.7	186.4	190.9	194.4	192.2	
	Temperatures (°F.):												
9	Furnace				2554	2617	2350	2637	2568	2457	2497	2380	
10	Gas uptake	664.8	541.0	462.8	535.9	518.4	467.7	534.7	522.7	495.2	527.3	473.9	
11	Feed water	207.3	187.3	200.8	187.3	176.1	187.2	193.4	198.4	180.9	187.5	176.9	
12	Superheated steam	645.3	466.7	441.6	462.8	455.1	445.8	461.5	459.9	452.3	464.0	441.3	
13	Calorimeter superheater inlet	311.5	311.8	310.8	310.6	310.7	310.8	311.0	310.0	311.0	311.7	310.3	
14	Room	106.9	109.6	99.6	96.1	95.9	88.5	96.3	86.9	90.3	101.2	95.4	
15	Air entering stoker	94.1	95.6	90.8	111.5	111.5	109.7	113.6	105.2	114.3	134.3	117.8	
	Air pressures (in H_2O):												
16	Main duct	2.39	3.52	2.97	3.54	3.90	2.37	2.97	3.25	3.65	3.17	1.99	
17	Compartment No. 1	0.057	0.217	0.179	0.179	0.139	0.100	0.128	0.141	0.231	0.369	0.228	
18	Compartment No. 2	0.319	0.636	0.280	0.776	0.541	0.459	0.616	0.637	0.736	1.145	0.701	
19	Compartment No. 3	0.158	0.386	0.155	0.444	0.304	0.419	0.363	0.316	0.664	0.987	0.637	
20	Compartment No. 4	−.006	−.075	−.072	−.012	0.000	0.014	0.000	0.000	0.334	0.497	0.351	
21	Air duct over fire	0.525	1.917	0.575	2.139	1.557	0.300	1.782	1.958	0.700	0.419	0.137	
	Drafts (in. H_2O):												
22	Furnace	0.153	0.203	0.176	0.139	0.147	0.087	0.093	0.121	0.058	0.061	0.062	
23	Uptake	0.630	1.349	0.612	1.259	1.038	0.598	1 002	1.001	0.992	1.377	0.730	
	Gas analysis (%):												
24	Top 1st Pass. CO_2	15.06	14.57	15.16	13.71	15.89	16.22	16.11	15.92	14.12	14.78	13.76	
25	Top 1st Pass. O	3.25	3.31	4.15	7.45	2.76	4.29	3.29	3.96	6.48	4.15	5.24	
26	Top 1st Pass. CO	0.01	0.69	0.00	0.07	0.13	0.00	0.08	0.07	0.00	0.00	0.00	
27	Top 1st Pass. N	81.68	81.31	80.75	81.18	81.22	81.49	81.61	81.11	81.15	81.07	81.02	
28	Top 3rd Pass. CO_2	13.07	13.04	13.32	13.22	13.65	13.08	14.50	13.74	12.98	14.61	13.11	
29	Top 3rd Pass. O	6.14	5.25	6.16	5.23	5.35	5.61	4.52	5.15	6.17	4.16	5.94	
30	Top 3rd Pass. CO	0.00	0.00	0.00	0.33	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
31	Top 3rd Pass. N	80.79	80.65	80.65	81.16	80.70	81.35	81.28	81.11	80.95	81.29	80.85	
32	Quality of smoke	0 to .25	.50	Trace	.50	.25	0 to .25	.25	0 to .25	Trace	0	0	
33	Total water evaporated (lbs.)	61245	83109	60853	75808	76305	61064	79954	73391	86114	101608	67908	
34	Total wet coal fired (lbs.)	6989	9917	6946	9209	8840	7191	9009	8366	8772	10484	6826	
35	Total refuse, dry (lbs.)	1016.9	1205.3	877.0	1165.1	1266.0	1037.0	1084.1	913	760	967	580	
36	Moisture in coal (%)	8.58	8.42	7.60	11.18	7.07	7.31	7.23	8.06	4.30	3.21	2.74	
37	Quality of steam S.H. index (%)	99.90	99.88	99.90	99.90	99.88	99.87	99.87	99.87	99.89	99.90	99.88	
38	Calculated saturated steam temperature at S. H. outlet (°F.)	385.76	386.52	383.10	382.38	383.60	384.92	385.36	382.36	384.24	385.64	384.68	
39	Degrees superheat at S. H. outlet (°F.)	59.34	80.18	58.50	80.42	71.50	60.88	76.14	77.56	68.06	78.36	57.62	
40	Factor of evaporation	1.0040	1.1253	1.0987	1.1248	1.1318	1.1146	1.1171	1.1179	1.1251	1.1245	1.1229	
41	Total water F & A. 212°F. (lbs.)	66965.3	93539.2	66909.8	85248.8	86398.0	68061.9	89316.6	82301.1	96886.9	114238.2	76253.8	
42	Water evaporated per hr. actual (lbs.)	10207.5	16621.8	10050.8	15161.6	12717.5	9294.0	13326.7	12245.2	13248.5	10864.7	11318.0	
43	Water per hour F & A. 212°F. (lbs.)	11160.9	18707.8	11081.6	17053.8	14398.7	10471.1	14886.1	13700.2	14905.7	12043.0	12709.6	
44	Water per hour F & A. 212°F. per sq. ft. of H. S. (lbs.)	4.125	6.918	4.094	6.302	5.320	3.870	5.500	5.062	5.508	7.667	4.697	
45	Total dry coal (lbs.)	6261.9	9082.0	6438.6	8362.2	8215.0	6665.3	8249.6	7670.1	8284.8	10137.1	6648.7	
46	Wet coal per hour (lbs.)	1165.2	1983.4	1361.3	1841.8	1473.3	1106.2	1501.5	1436.7	1349.5	1748.0	1138.3	
47	Dry coal per hour (lbs.)	1069.3	1816.4	1073.1	1636.4	1369.2	1025.4	1374.9	1311.7	1281.5	1682.9	1108.1	
48	Dry coal burned per cu. ft. of F. V. per hour (lbs.)	6.38	10.86	6.39	9.80	8.20	6.14	8.23	7.85	7.72	10.16	6.64	
49	Dry coal burned per sq. ft. of H. S. per hour (lbs.)	0.393	0.671	0.397	0.605	0.506	0.379	0.314	0.485	0.477	0.626	0.409	

45	Total dry coal (lbs.)	6351.9	9082.0	6438.6	8862.2	8215.0	6645.3	8249.8	7870.1	8894.8	10157.1	6648.7
46	Wet coal per hour (lbs.)	1139.8	1883.4	1161.3	1841.9	1478.3	1198.0	1501.3	1436.7	1349.3	1749.6	1139.3
47	Dry coal per hour (lbs.)	1088.3	1816.6	1073.1	1636.4	1369.2	1025.4	1374.6	1311.7	1291.5	1692.9	1189.1
48	Dry coal burned per sq. ft. of P. V. per hour (lbs.)	6.38	10.80	6.43	9.80	8.20	6.14	8.33	7.85	7.73	10.14	6.64
49	Dry coal burned per sq. ft. of H. S. per hour (lbs.)	0.392	0.671	0.397	0.605	0.506	0.379	0.514	0.485	0.477	0.626	0.409
50	Wet coal burned per sq. ft. of grate surface per hour (lbs.)	20.00	34.20	20.02	32.89	25.40	19.07	26.81	24.89	23.27	30.15	19.64
51	Dry coal burned per sq. ft. of grate surface per hour (lbs.)	19.28	31.32	18.50	28.21	23.61	17.68	23.99	23.62	22.27	29.19	19.11
52	Actual evaporation per lb. dry coal (lbs.)	9.63	9.15	9.37	9.27	9.29	9.16	9.52	9.38	10.24	10.08	10.21
53	Equivalent evaporation per lb. dry coal (lbs.)	10.53	10.30	10.38	10.43	10.51	10.21	10.71	10.44	11.34	11.31	11.45
54	B. t. u. per lb. dry coal	13072	13277	12867	13180	13208	12875	13190	13225	14443	14548	14572
55	B. t. u. liberated per cu. ft. P. V. per hour	83994.1	144413.9	82734.8	129180.8	108388.2	79053.5	110152.2	103845.6	113183.7	147516.7	96758.1
56	Total air supplied per hour (lbs)	12730.0	21633.2	13381.6	18962.2	16186.7	12854.5	16866.9	15643.3	19750.9	25837.7	16837.1
57	Per cent excess air used for combustion	17.75	18.01	24.17	12.80	14.91	11.57	15.67	13.91	24.05	32.45	26.40
58	Average Horsepower per hour—stoker				0.706	0.706	0.692	0.699	0.709	0.707	0.699	0.708
59	Average Horsepower per hour—ash conveyor				0.247	0.206	0.248	0.184	0.233	0.182	0.256	0.174
60	Average total H. P. per hour—stoker and ash conveyor				0.951	0.912	0.941	0.883	1.042	0.889	0.954	0.882
61	Efficiency—boiler, superheater and furnace (%)	78.17	75.28	77.68	76.72	77.23	76.95	78.01	76.62	76.48	78.64	76.38
	Heat balance (%):											
62	Heat absorbed by boiler	78.17	75.28	77.68	76.72	77.23	76.95	78.01	76.62	76.48	78.64	76.38
63	Loss due to moisture in coal	0.90	0.85	0.77	1.19	0.71	0.75	0.73	0.83	0.36	0.28	0.23
64	Loss due to water from combustion of hydrogen	3.88	3.93	3.94	4.19	4.12	4.17	4.10	4.15	3.39	3.42	3.36
65	Loss due to dry chimney gases	9.14	10.12	9.07	11.24	10.03	9.78	9.84	10.43	10.80	10.13	9.92
66	Loss due to incomplete combustion of carbon	0.00	1.90	0.00	1.35	0.00	0.00	0.00	0.00	0.00	0.00	0.00
67	Loss due to unburned carbon in refuse	2.61	1.33	2.33	2.74	3.34	3.81	2.72	2.96	2.71	3.43	3.39
68	Radiation and unaccounted for	5.28	7.43	6.21	2.57	4.57	4.54	4.60	5.71	6.24	7.30	6.72
69	Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
70	Refuse—per cent of dry fuel	15.98	13.27	13.62	14.24	13.41	15.60	12.03	11.99	8.63	9.72	8.73
	Refuse analysis:											
71	Combustible	18.48	13.65	15.09	17.35	21.07	21.57	20.38	16.25	24.18	25.14	23.78
72	Ash	85.52	86.35	84.91	82.65	78.93	78.43	79.63	83.75	65.82	64.86	61.22

COAL ANALYSIS

	ULTIMATE ANALYSIS OF DRY COAL Dundon gas sleek Dundon W. Va.	Pocahontas Screened coal West Virginia
Carbon	74.47	86.97
Hydrogen	4.90	4.49
Nitrogen	1.42	1.24
Sulphur	.66	.68
Ash	11.94	5.74
Oxygen	6.61	1.88
PROXIMATE ANALYSIS		
Moisture	7.36	2.41
Volatile matter	33.65	18.40
Fixed carbon	54.21	73.86
Ash	11.94	5.74
Sulphur	.66	.68
Fusing point of ash(°F)#	2600	2410

Defined as softening temperature in A. S. T. M. Specification D-32-24.

THE EFFECT OF DISARMAMENT AND TREATY LIMITS UPON NAVAL ENGINEERING.

By Lieutenant Carl J. Lamb, U. S. N. R., Member.

Prior to the Washington treaty, and actually until some time after its ratification, little or no serious thought had been given by Naval designers toward obtaining light weight machinery for propelling Naval vessels, other than submarines and destroyers. Practically all our battleships were propelled by reciprocating engine, direct-drive turbine or turbine-electric machinery; scout cruisers and destroyers by cross-compound geared turbines, having cruising belts or cruising turbines; submarines by the dual Diesel-battery system; and various auxiliary vessels by either reciprocating engines or cross-compound geared turbines.

As far as existing ships are concerned, the situation has not materially changed, except to a slight extent, as a result of modernization, but due to the ever-important necessity of obtaining the most efficient possible fighting ships within displacement tonnage limits prescribed by existing treaties, vessels that are to be built in the future will be found to have engineering plants which are great improvements over anything employed heretofore in Naval engineering practice.

An examination of Table X shows the average total weight per S.H.P. of various types of vessels and different types of propelling machinery, as found in the U. S. Navy. Companion Table Y shows the total weights and unit weight per S.H.P. of the same types of vessels, if propelled in each case by modern triple-divided-flow geared turbines with underneath condensers, and having modern auxiliaries. While it may be stated that machinery weights are known only for the 10,000 ton cruisers with the modern triple-divided-flow turbines, condensers, auxiliaries, auxiliary turbine-generators and blowers, the builders of such modern Naval propelling machinery have made weight calculations and studies for the various other types of vessels listed.

The different types of main geared turbines used to propel Naval and merchant vessels throughout the world, as a result of long experience and practice, have gradually demonstrated their fitness, or lack of such, to meet the exacting requirements of the duty involved, so that one particular type stands out above all the rest and is generally recognized as the most suitable for propelling sea-going ships. The impulse-reaction turbine is considered by the majority of marine and Naval engineers today, the most reliable, simple, suitable, and over an extended period, the most efficient possible of development for merchant or Naval work.

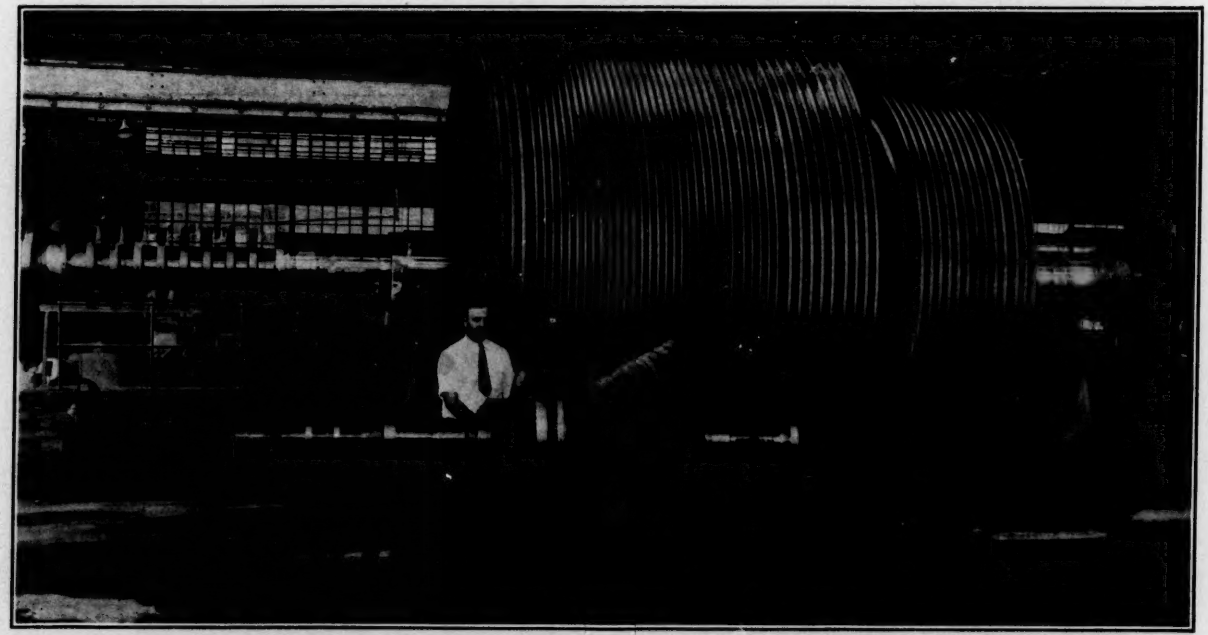
Until limitations were set upon displacement tonnage by treaty, no effort had been made to reduce appreciably the weight per S.H.P. of the entire machinery plant, because it was commonly considered that sturdiness and reliability were necessarily accompanied by ample weight and size. This fact may be appreciated by examining Table X.

It is interesting to see what advances were being made by experienced specialized builders, in the design of central station turbines, with regard to weights, during a corresponding period of time, as may be noted in Table Z.

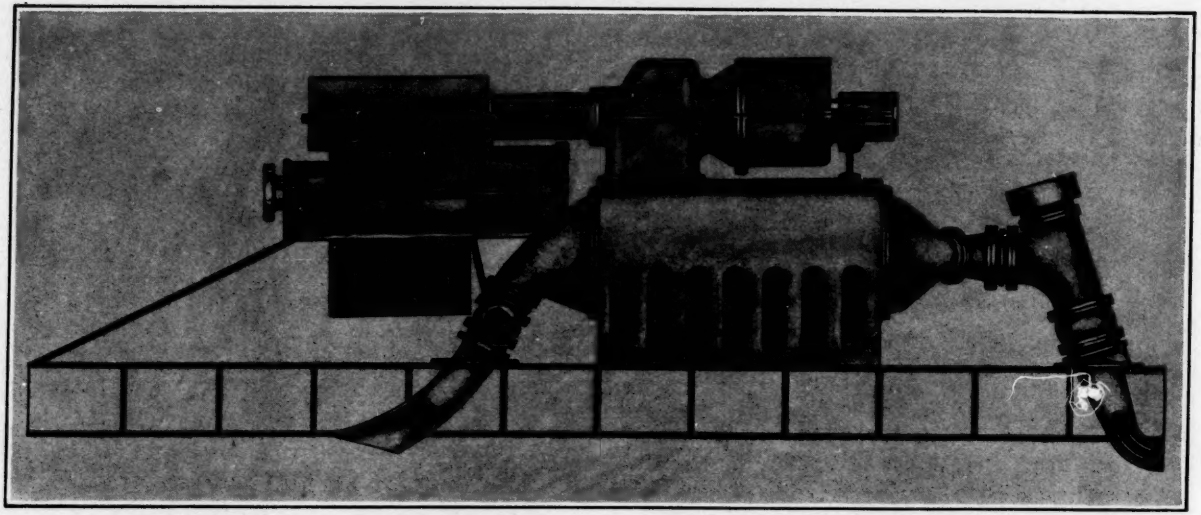
Some short time ago, an interesting remark was made by a high ranking officer of another Navy to the effect that the various limitations treaties had forcefully brought out one important point as far as engineering practice in that particular Navy was concerned, the point being that while great efforts had been made to obtain more efficient boilers and geared turbines, little constructive thought had been given toward improving the efficiency of all-important auxiliaries, or toward general weight reduction.

While the above was probably true to a great extent, in practically all navies, certain people in this country, encouraged by the Bureau of Engineering, had long been working toward the perfection of distinctly new types of main propelling turbines and the more important engine and fireroom auxiliaries. As a matter of fact, it was over fifteen years ago that Mr. H. F. Schmidt * invented the divided-flow turbine for Naval use, and had also started work on his propeller pump and blower experiments. Although the first divided-flow turbines for Naval use were

* See article on the Swedish Cruisers by W. W. Smith, August, 1917, A. S. N. E. JOURNAL.



OLD 7000 S.H.P. DIRECT CONNECTED, 220 R.P.M., BATTLESHIP TURBINE SPINDLE WITH MODERN TURBINE SPINDLE OF THE SAME HORSEPOWER BUT 3600 R.P.M., IN THE FOREGROUND.



NEW TYPE NAVAL GEARED TURBINE WITH UNDERNEATH CONDENSER AND NEW TYPE SCOOP.

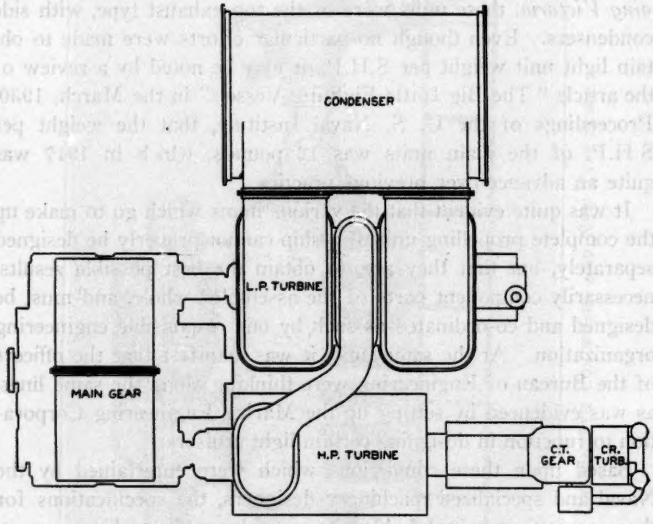
installed in the Swedish Baby Battleships, *Gustav V* and *Drottning Victoria,* these units were of the top exhaust type, with side condensers. Even though no particular efforts were made to obtain light unit weight per S.H.P., it may be noted by a review of the article "The Big Little Fighting Vessels" in the March, 1930, Proceedings of the U. S. Naval Institute, that the weight per S.H.P. of the main units was 12 pounds, which in 1917 was quite an advance over previous practice.

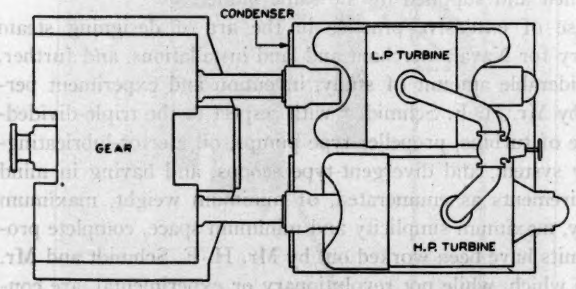
It was quite evident that the various items which go to make up the complete propelling unit of a ship cannot properly be designed separately, but that they are, to obtain the best possible results, necessarily component parts of the assembled whole, and must be designed and co-ordinated as such by one responsible engineering organization. At the same time, it was manifest that the officers of the Bureau of Engineering were thinking along the same lines, as was evidenced by setting up the Marine Engineering Corporation to function in designing certain light cruisers.

Based upon these convictions which were entertained by the Naval and specialized machinery designers, the specifications for three cruisers required bidders to consider main turbines, gears, thrust bearing, condensers, condenser auxiliaries, oil pumps, oil coolers, scoops and piping as one complete propelling unit, all to be designed and supplied by the same bidder.

Because of extensive practice in the art of designing steam machinery for Naval, merchant and land installations, and further, the considerable amount of study, invention and experiment performed by Mr. H. F. Schmidt * with respect to the triple-divided-flow type of turbine, propeller-type pumps, oil ejector-lubricating-governor system, and divergent-type scoops, and having in mind the requirements as enumerated, of minimum weight, maximum reliability, maximum simplicity and minimum space, complete propelling units have been worked out by Mr. H. F. Schmidt and Mr. Flanders which, while not revolutionary or experimental, are considered to indicate great advance in the art of Naval machinery design, and to mark the beginning of a new era in U. S. Naval engineering.

* See JOURNAL A. S. N. E. Prize essays, 1927 and 1929, also, May, 1929, issue of "Marine Engineering and Shipping Age."





COMPARATIVE SKETCHES OF CROSS-COMPOUND AND TRIPLE DIVIDED FLOW UNITS - SAME SCALE

COMPARATIVE SKETCHES TO SAME SCALE OF OLD CONVENTIONAL TYPE OF CROSS-COMPOUND GEARED TURBINE, AND NEW TYPE, BOTH BEING SAME HORSEPOWER AND R.P.M.

COMPARATIVE SKETCHES TO SAME SCALE OF OLD AND NEW TYPES OF CONDENSING, GEARED, 250 KW., TURBO-GENERATOR SETS.

The units being discussed consist essentially of an impulse-reaction high pressure condensing turbine mounted beside an impulse-reaction medium pressure double flow condensing turbine, both being supported by an underneath radial-flow condenser, having two tube nests. The turbines drive the main gear wheel through pinions located on its upper quarters. A cross-over pipe runs from the impulse-chamber of the high pressure to the nozzle block of the medium pressure turbine. The steam flow for cruising and for all speeds up to 25 knots is through the high pressure turbine only, and then downward to the condenser. For speeds above 25 knots and up to full power, all steam flows through the impulse wheel of the high-pressure turbine, approximately 2/9 of the steam going on through the reaction section of the same turbine to the condenser, and 7/9 going from the impulse chamber through the cross-over pipe to the double-flow low-pressure turbine and thence downward to the condenser. This triple-divided-flow design thus assures that at partial loads, only one turbine is doing the work, thereby obtaining an efficiency which results in an almost flat water rate curve, being comparable to that obtained with the cross-compound cruising turbine combination, where the work is divided among three turbines for the same power, while for powers from 25 knots up to full speed the work is evenly divided between the two turbines, there being no idle blades or turbines in service.

This combination of turbines and condenser eliminates exhaust trunks, expansion joints, drain pipes, and, since the turbine weights are entirely supported by the condenser, greatly reduces foundation weights (and costs). Because the condenser is served by a divergent-type scoop, having the standby propeller-type circulating pump constructed as an elbow in the inlet water piping, only one ship frame is pierced by each scoop pipe, no foundations, by-pass valves nor piping being required. Here again a considerable weight saving is gained, even of sea water in the system. The vacuum apparatus consists of vertical condensate pumps and condensing two-stage steam jet air ejectors.

The Schmidt oil-governor ejector-lubricating system has been employed, the ahead and astern maneuvering valves (which are cast in one piece with the steam strainer) being under oil pressure

from the governor impeller at all speeds. The basic characteristic of the main oil impeller is that the pressure varies as the square of the speed, so that once a certain speed has been set by the maneuvering valve, the turbine will be kept steady at that speed, even though the steam pressure should vary. Since all of the high-pressure oil delivered by the main oil impeller, which is located on the turbine shaft, is not required for governor action, part of it is discharged as motive oil through an ejector (which takes suction from the oil sump tanks) to the bearing lubricating system, thus obtaining complete lubrication simply and reliably at the main turbine water rate, rather than at the inefficient rate of small separate independent pumps. Each main gear case serves as the oil sump tank, to which are secured the oil ejector and two small two-stage vertical propeller-type auxiliary oil pumps to furnish governor priming and lubricating oil for starting, stopping, maneuvering and speeds below 10 knots.

The oil cooler, instead of being of two-pass flow supplied with cooling water by a separate pump, is single-pass, mounted on or close to its main condenser, taking water from the main injection scoop, and discharging to the main overboard water box, so that a further saving in weight of piping and water has been secured, and greater simplicity, of course.

After a very careful study and analysis of the proper materials available, selection was made of those which would result in maximum strength with minimum weight. All turbine rotors are solid steel forgings; blades, stainless steel; turbine cylinders, cast steel; gear cases, boiler plate steel; gear centers, steel forgings, and condenser shells, plate steel.

From the general, unillustrated description given above, it can be concluded that a main propelling unit so designed, while effecting great saving of space, weight, foundations and piping, should be extremely sturdy and, because of greater accessibility and simplicity, of greater reliability than anything used heretofore in Naval practice. It is quite evident that, although costs were not the basic consideration, such machinery should reduce installation costs of foundations, piping, platforms, etc., as generally occurs when greater simplicity is obtained.

During the same time that the above design work has been going on to produce such a co-ordinated main propelling unit, the develment of the Schmidt propeller-type blower * had reached a stage, especially with respect to higher efficiency, utmost simplicity and greater reliability, than at first seemed possible, that this type of blower has been purchased to supply boiler room forced draft on the same three cruisers which will be propelled by the triple-divided-flow turbines. See Table W.

While there are, of course, various opinions and viewpoints as to the effects upon our Navy of the limits set with regard to numbers and tonnage of the various classes of Naval vessels, there can be no doubt but that such treaty limitations have resulted in study aimed toward the development of greatly improved machinery plants. Aside from the weight savings effected by the adoption of the complete units described above, a careful recapitulation shows the following very important advantages which indicate the selection of such units for future U. S. Navy vessels:

1. Unified and complete design responsibility.
2. Greatest possible simplicity, with resultant greater reliability.
3. Greater accessibility, and less space required per unit S.H.P.
4. Greater overall fuel economy.
5. Elimination of cruising turbines, gears, and clutches; also of exhaust trunks, considerable foundations and a large amount of piping and valves.
6. Underneath condensers insure perfect drainage of turbines at all times, without use of drain piping, valve ejectors, or pumps.
7. High-pressure turbines, only, in use for cruising and up to 25 knots; above 25 knots and at full power, no idle blading in service.
8. All essential auxiliaries designed to be component parts of the main propelling machinery.
9. The use of the simplest, most reliable and efficient lubricating system devised to date.
10. Considerable gaining of strength and saving of weight and cost because of divergent scoops, since only one hull frame is pierced.

* See an article by Messrs. Lincoln and Treat, JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, August, 1928, issue.

11. Ability, because of the nature of the design, to obtain approximately 75 per cent of full power on either one of each pair of turbines, in case of battle casualty.

12. Better protection for all main machinery from gunfire, since it is located well below the water line.

WEIGHT TABLE W, CERTAIN AUXILIARIES.

Name	Weight Old Design	Weight New Design	Capacity
	Pounds	Pounds	
Forced-draft blowers	3400	1900	75,000 cu. ft. 10″ S. P.
Aux. geared D. C. Turbine-Gen.	20,000	11,000	250-Kw.
Circulating pumps (turbine-driven)	10,000	4000	15′ head 15,000 G.P.M.
Condensate pumps (turbine-driven)	6000	3800	60′ head 200 G.P.M.
Oil pumps, turbine-driven	5000	1000	70′ head 400 G.P.M.
Air ejectors, 2-stage, condensing	4000	2000	

TABLE "X"

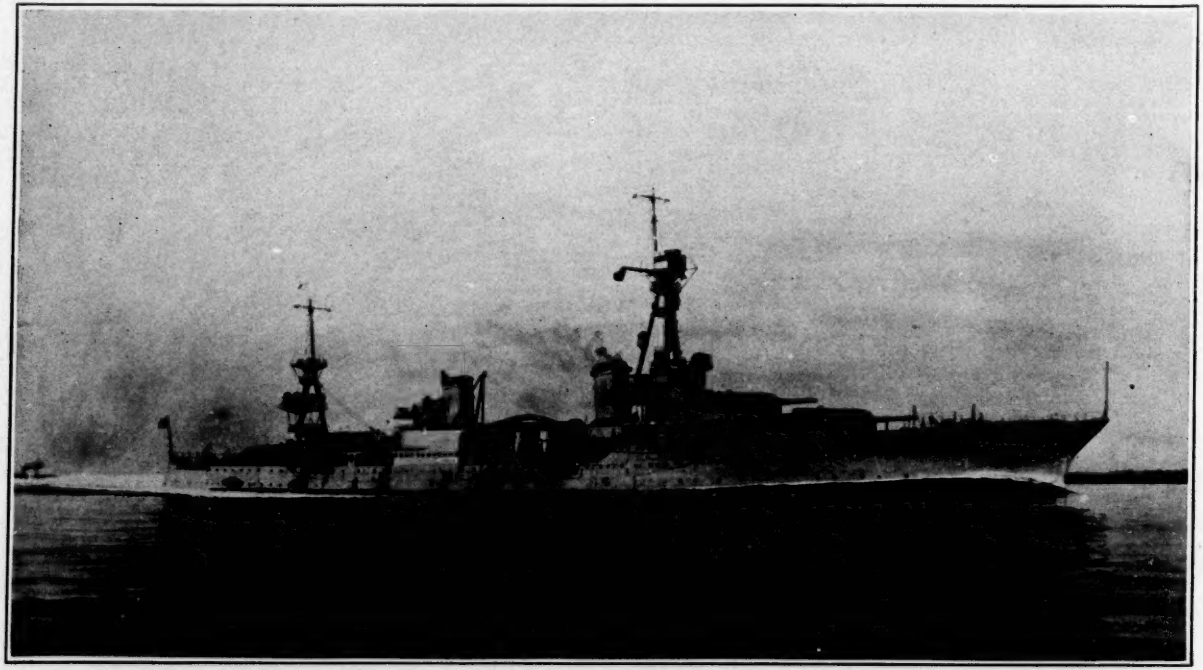
Ship	Type of Drive	Date Accepted	Total S.H.P.	Total Wet Weight, all Machinery	Weight per S.H.P.
				Tons	
Battleship	Reciprocating	1914	28,373	2271	180
Battleship	Direct-drive turbine	1919	33,100	2703	183
Battleship	Turbine-electric	1920	29,609	2045	155
7500-ton cruiser	Cross-comp. geared-turbine with cruising turbines	1923	94,920	1722	41
10,000-ton cruiser	Cross-comp. geared-turbine with cruising turbines	1930	106,750	2161	45 Est.
1250-ton destroyer	Cross-comp. geared-turbine with cruising turbines	1919	28,000	450	36
Airplane	Turbine-electric	1928	180,000	7075	88 Est.

TABLE "Y"

Ship	Type of Drive	Total S.H.P.	Total Wet Weight all machinery in plant on ship	Weight per S.H.P.
			Tons	Estimated
Battleship	Triple-divided-flow turbines	33,100	1108	75
7500-ton cruiser	Triple-divided-flow turbines	94,000	1425	34
10,000-ton cruiser	Triple-divided-flow turbines	106,750	1820	38
1250-ton destroyer	Triple-divided-flow turbines	28,000	375	30
Airplane carrier (*Saratoga* type)	Triple-divided-flow turbines	180,000	2500	31
Airplane carrier (new Type contracted for)	Triple-divided-flow turbines	53,500	960	40

TABLE "Z"

	Date	Rating	Total Weight of Turbine	Speed	Weight per B.H.P.
		Kilowatts	Pounds	R.P.M.	Pounds
Plant "A"	1914	60,000	1,600,000	1200	16.0
Plant "B"	1928	65,000	600,000	1800	5.6
Plant "C"	1915	12,000	175,000	1800	8.7
Plant "D"	1929	12,000	89,000	3600	4.42



U. S. S. "NORTHAMPTON."

U.S.S. *NORTHAMPTON*, U.S.S. *CHESTER*, U.S.S. *HOUSTON*.

Construction, Description, and Official Preliminary Acceptance Trial Data.

By Lieut. Comdr. J. K. Esler, U. S. Navy, Member.

Of the eight light cruisers authorized by Congress in an act approved 18 December, 1924, the U.S.S. *Pensacola* (CL24) and the U.S.S. *Salt Lake City* (CL25), described in this Journal in Volume XLII, No. 2, May, 1930, were the first to be constructed.

Contracts for four of the remaining six vessels were awarded as follows:

CL26 (*Northampton*) to Bethlehem Shipbuilding Corporation of Quincy, Mass., on 13 June, 1927, due for delivery 13 June, 1930.

CL27 (*Chester*) to New York Shipbuilding Co., on 13 June, 1927, due for delivery 13 June, 1930.

CL30 (*Houston*) to Newport News Shipbuilding and Dry Dock Corporation, on 13 June, 1927, due for delivery 13 June, 1930.

CL31 (*Augusta*) to Newport News Shipbuilding and Dry Dock Co. on 13 June, 1927, due for delivery 13 March, 1931.

Plans for these four vessels were prepared by the Marine Engineering Corporation, and were taken in part from plans already prepared for the U.S.S. *Salt Lake City* (CL25) and the U.S.S. *Pensacola* (CL24). Insofar as possible at the advanced stage of construction the plans for these latter vessels were developed in conjunction with the plans for the other four. Plans also were furnished to the Navy Yards at Puget Sound and at Mare Island to assist in the construction of the U.S.S. *Louisville* (CL28) and the U.S.S. *Chicago* (CL29). As a result of this centralized design, the machinery installations of the *Northampton*, *Chester*, *Houston*, and *Augusta* are practically identical, and the engineer-

ing plants of the other four vesesls are similar in general features of arrangement and design.

Dates of trials, delivery to the Government, and commissioning are as follows:

	Prelim. Trials	*Delivery*	*Commissioned*	*Final Trials*
U.S.S. *Pensacola*	Built at Navy Yard	(No preliminary Trials)	6 Feb. 1930	28-29 July 1930
U.S.S. *Salt Lake City*	12-19 Nov. 1929	9 Dec. 1929	11 Dec. 1929	12-13 May 1930
U.S.S. *Northampton*	15-21 April 1930	15 May 1930	17 May 1930	21 Oct. 1930
U.S.S. *Chester*	20-25 May 1930	23 June 1930	24 June 1930	Scheduled 2 Dec. 1930
U.S.S. *Houston*	3-7 June 1930	17 June 1930	17 June 1930	Scheduled 18 Nov. 1930
U.S.S. *Augusta*	Due for delivery 13 March 1931			
U.S.S. *Chicago*	Due for delivery 13 March 1931			
U.S.S. *Louisville*	Due for delivery 13 March 1931			

GENERAL CHARACTERISTICS.

Length over all, feet	600
Breadth extreme, feet and inches	66–7/8
Displacement—Standard, tons	10,000
Designed speed, knots	32.7
Designed R.P.M.	366.0
Designed S.H.P.	107,000

The machinery installations on the U.S.S. *Northampton*, U.S.S. *Chester*, U.S.S. *Houston*, and U.S.S. *Augusta* are arranged as shown on accompanying folder. Description of machinery is as follows:

MAIN ENGINES.

Four—Main turbines, Parsons, geared; one H.P., L.P. and cruising turbine operating through single reduction gears on each shaft. Astern turbines—all impulse blading—are located in the L.P. turbine casing. Turbine design in general follows the design on the *Richmond* class light cruisers, the principal departure

being a cruising turbine for each unit. Each unit is designed to develop continuously 26,750 shaft horsepower. The cruising turbine is designed for operation through single reduction gearing and Metten hydraulic clutch to H.P. turbine shaft at speeds up to about 20 knots and to operate disconnected from main turbines at all times if desired, in which case it acts merely to drive the connected pumps. When operating independently, the cruising turbine is governor controlled at a speed corresponding to about fifteen knots. Automatic operation of exhaust valves is provided for shifting cruising turbine exhaust from condenser to H.P. turbine when coupling, and from H.P. turbine to main condenser when uncoupling. Provision is made for bleeding steam from appropriate stages of the H.P. turbine to the auxiliary exhaust for use in feed water heaters and evaporators, and for using auxiliary exhaust in turbines if in excess. The number of electric motor driven auxiliaries make it improbable that there will be any excess auxiliary exhaust steam under any condition of cruising. Each cruising turbine is fitted to drive main air pump, lubricating oil pump and turbine drain pump. Additional pumps which were, in the original installation, connected for use as cruising feed pumps, found unsatisfactory for that purpose and retained for use as make up feed and bilge pumps, were removed from these vessels soon after and as a result of their performance on the preliminary acceptance trials. Independent reciprocating pumps 7 inches × 7 inches × 12 inches have been installed, one in each engine room as make up feed pumps. Direct connected pumps driven by cruising turbines, are as follows:

Main Air Pump—Edwards type two cylinder.
Lubricating Oil Pump—Quinby Screw Type.
Turbine Drain Pump—Special solid plunger single acting pump.

At cruising speeds the main air pump suction lines in each engine room may be cross connected, thus permitting the operation of but one cruising turbine under this condition if desired, and also providing that one cruising turbine air pump may take condensate from both condensers in case of the failure of one cruising turbine unit.

The after engine room is the control engine room.

BOILERS.

8—Babcock and Wilcox White Forster, Oil burning, arranged in four firerooms.

Furnace volume—one boiler—1137 cubic feet.

Heating surface—one boiler—11,880 square feet.

No. burners—14—per boiler—Cuyama type.

Steam pressure—working—300 pounds gauge.

Authorized safety valve setting 310 pounds.

Steam—not superheated.

Soot blowers—Diamond type.

Periscopic smoke indicators—2 per boiler.

Burner telegraph—operated from after engine room.

Boiler safety valves—two duplex 4-inch on each boiler. Form E. Consolidated Ashcroft Hancock Company, modified to include new design features.

PROPELLERS.

4—Solid, 3 blade, Manganese bronze.

Diameter—12 feet 0 inches.

Pitch—mean—11 feet 9 inches.

Heliocoidal area—72 square feet.

Projected area—62.2 square feet.

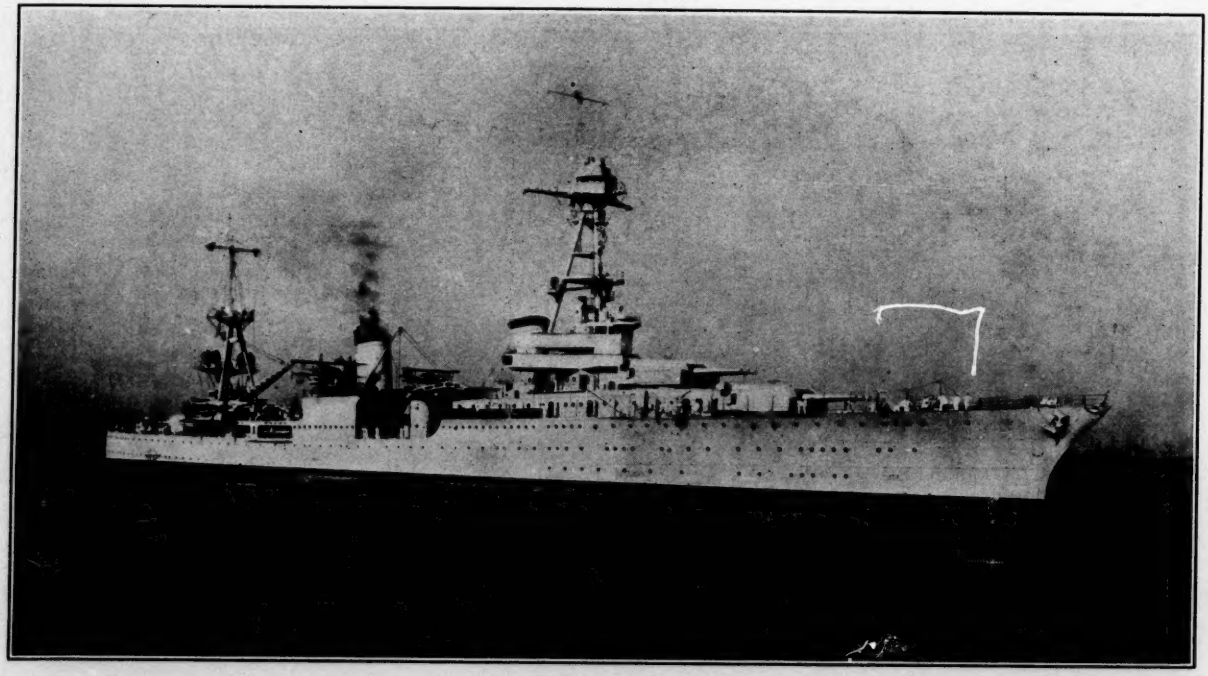
Designed for 366 R.P.M. at 107,000 S.H.P.

FORCED DRAFT BLOWERS.

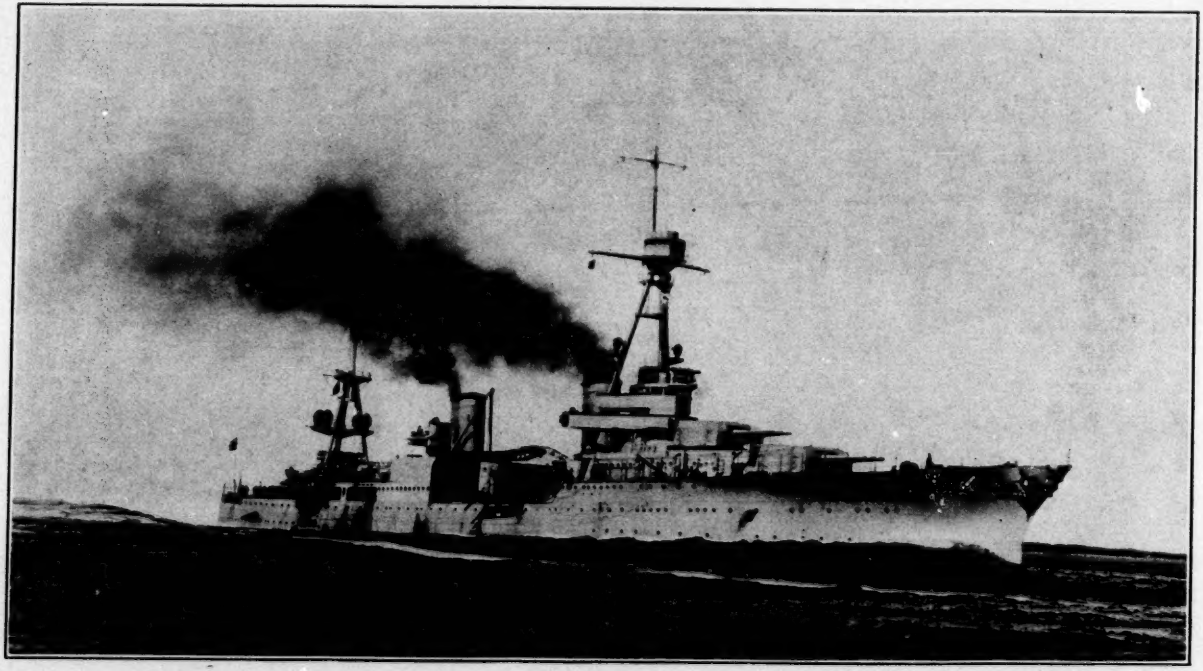
8—2 in each fireroom—Sturtevant—vertical—turbine driven—75,000 cubic feet of air per minute capacity each. Intake trunks are fitted with automatic spring balanced shutters. There is no arrangement on these vessels for utilizing engine room ventilation in firerooms. Escape and access trunks are led to floor plate level on each side of each fireroom.

MAIN GENERATORS.

4—General Electric three wire 250 kilowatt turbine driven through reduction gears. Two sets installed in each engine room. Turbines are designed to operate condensing only. One set is fitted with exhaust connection to auxiliary exhaust line for emergency use or when in dry dock. Turbine speed 10,012 R.P.M., generator speed 1200 R.P.M.



U. S. S. "HOUSTON."



U. S. S. "CHESTER."

The two sets in each engine room are located directly above a dynamo condenser, which serves also as an auxiliary condenser. Dynamo air and circulating pumps are motor driven, the air pump being reciprocating and the circulating pump centrifugal. Connections are provided for draining condenser and air pump to reserve feed tanks and also for use of dynamo air pumps in drying out main and cruising turbines through turbine drains.

MAIN CONDENSERS.

4—One for each main propelling unit. Uniflow type with scoop injection and turbine driven direct connected vertical main circulating pumps for use when stopped, backing, or maneuvering at low speeds. The throttle for these pumps is located at the working platform and pump turbine exhaust valve is fitted with steam piston operated disk to open the exhaust valve automatically when pump throttle is opened. Main condenser cooling surface 12,692 square feet each. Absolute pressure gauges (mercury) are fitted on each main condenser in addition to the dial type spring actuated gauges. Augmentors and augmentor condensers are installed.

MAIN FEED PUMPS.

4—Bethlehem Weir—turbine driven, single stage, centrifugal, designed for 750 G.P.M. with 400 pounds gauge outlet pressure when operating at 5250 R.P.M. Two pumps are located in each engine room with direct suction from main feed tank. Pressure regulators are installed.

EMERGENCY FEED PUMPS.

4—Worthington, vertical, double acting 14 inches × 9 inches × 24 inches, one in each fire room, with suction to double bottom reserve feed tanks and discharge to boilers through or by passing the feed water heater in fireroom in which located.

FUEL OIL SERVICE PUMPS.

8—Quimby, vertical, rotary screw type, driven by Terry turbines, 2 in each fireroom.

Fuel oil tank heating system. Heating coils are fitted in all fuel

oil tanks in lieu of the recirculating system as installed on the U.S.S. *Salt Lake City*.

FUEL OIL BOOSTER PUMPS.

4—8 inches × 10 inches × 12 inches Vertical reciprocating double acting, single (Worthington) one in each engine room.

PORT OIL SERVICE PUMPS.

4—Rotary plunger type, motor driven, 15 G.P.M. capacity. One in each fireroom.

DISTILLING PLANT.

2—Double effect L.P. submerged tube evaporator sets, rated capacity, 20,000 gallons per day each plant.
2—Distiller condensers—horizontal type.
2—Tube nest drain heaters.
2—Vapor feed heaters.
2—Condensate coolers.
2—Turbine driven combination pumps, each with
1—reciprocating air pump 11 inches × 10 inches.
1—single suction volute distiller circulating pump, 400 G.P.M.
1—single cylinder brine overboard pump—single acting.
1—Sturtevant type A-17 turbine 1300 R.P.M.
1—Evaporator feed pump motor driven—90 G.P.M.
1—Meter—2 measuring tanks.

It is believed that except in emergency one set will supply all fresh water required.

REFRIGERATING PLANT.

2—Electric motor driven Kroeschell 2 ton Carbon dioxide compressors.
1—Carbon dioxide condenser.
1—Brine cooler.
2—Brine circulating pumps, centrifugal, motor driven.
2—Condenser circulating pumps, centrifugal, motor driven.
1—Ice making box, rated at 400 pounds per day.

It is believed that one machine will be sufficient for normal load. The Bureau of Construction and Repair have cognizance

over small individual refrigerating machines installed in Officer's and Chief Petty Officer's pantries. These are of the General Electric Type.

FIRE AND FLUSHING PUMPS.

2—Worthington centrifugal, provided with motor drive at one end of impeller shaft for use on flushing system and direct connected turbine drive on other end of impeller shaft for use as fire pumps. One pump is located in each engine room.

4—Fire and bilge pumps 10 inches × 9 inches × 12 inches reciprocating.

MACHINE TOOLS.

1—Gap lathe 16 inches × 32 inches × 8 feet extension bed.

1—Engine lathe 16 inches × 10 feet.

1—Universal Milling machine 28 inches × 10 inches × 18 inches.

1—Upright drill press—28 inches with sliding head.

1—Shaper 16 inches.

1—Sensitive drill 16 inches floor type.

1—Combined wet and dry wheel emery grinder.

1—Power hack saw.

All machine tools are fitted with independent motor drive.

The Bureaus propose to install an additional small lathe, tool grinder, and tool post grinder on each vessel of this class in accordance with recommendations of the trial board.

ELECTRICAL WORKSHOP.

A small electrical workshop is provided. As originally installed this consisted of a work bench with vises, a test panel, and lockers for stowage of instruments and tools. The portable battery charging panel and racks were located in this space. The portable battery charging panel and racks have been relocated in the searchlight rheostat room and the work bench extended. The Bureaus propose to install an armature bake oven and a coil winding machine in this space in accordance with recommendations of the trial board.

Electric Welding Outfit. Installation of electric welding outfits has been authorized for vessels of this class.

STORAGE BATTERIES.

Forward Battery Room.

General service battery—42 trays, 230 volt, 300 ampère hour.
Battle telephone battery—4 trays, 24 volt, 50 ampère hour.
Automatic telephone battery—8 trays, 48 volt, 50 ampère hour.
General Announcing battery—2 trays, 12 volt, 50 ampère hour.

After Battery Room.

General service battery—42 trays, 230 volt, 300 ampère hour.

Forward and after general service batteries are not interconnected.

X-ray equipment—An X-ray outfit has been authorized with outlets in sick bay.

SEARCHLIGHTS.

4—36-inch high intensity.
2—24-inch light weight high intensity, signal.
4—1000 watt incandescent, signal.

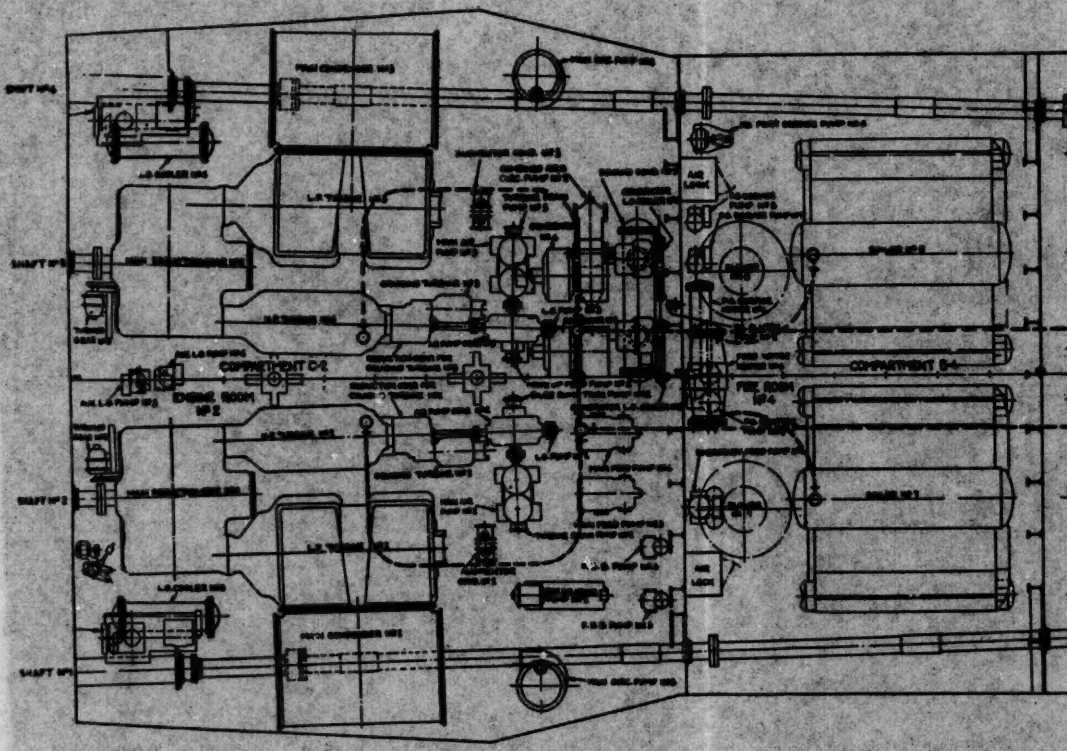
GYRO COMPASS EQUIPMENT.

2—Gyro compasses Arma Mark IV Mod. I are installed, one forward and one aft, with connections for use of forward and after storage batteries.

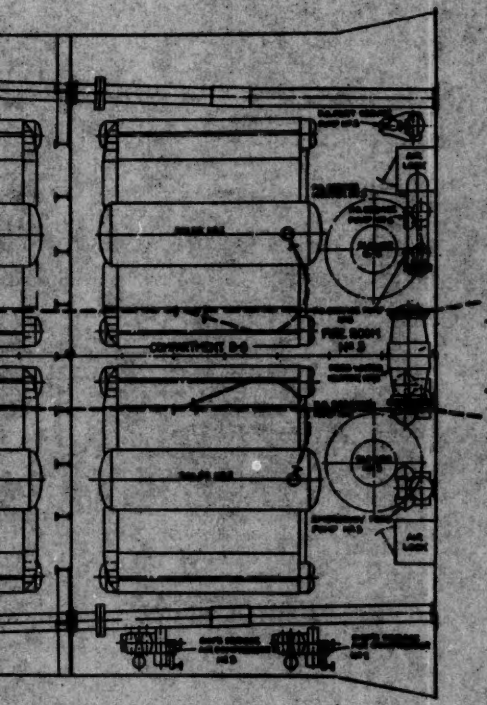
A comparison of data taken during Preliminary Acceptance Trials of the four vessels on which such trials have been conducted is shown in appended tables.

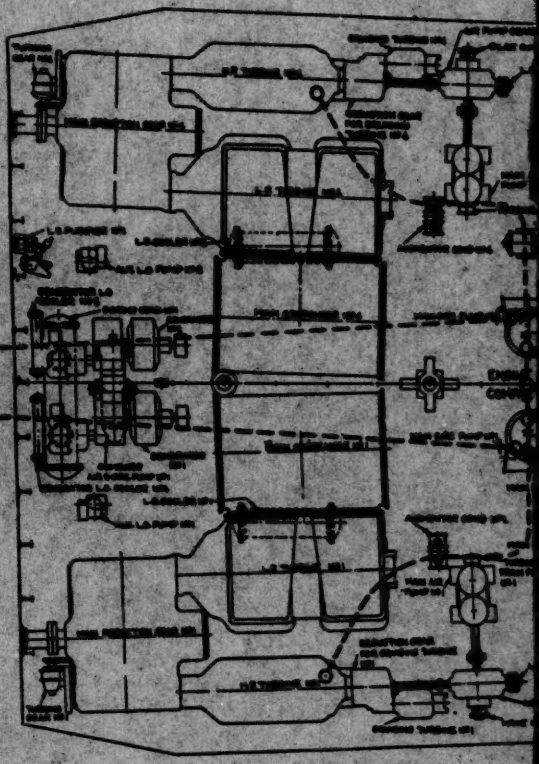
DATA TAKEN ON STANDARDIZATION RUNS.

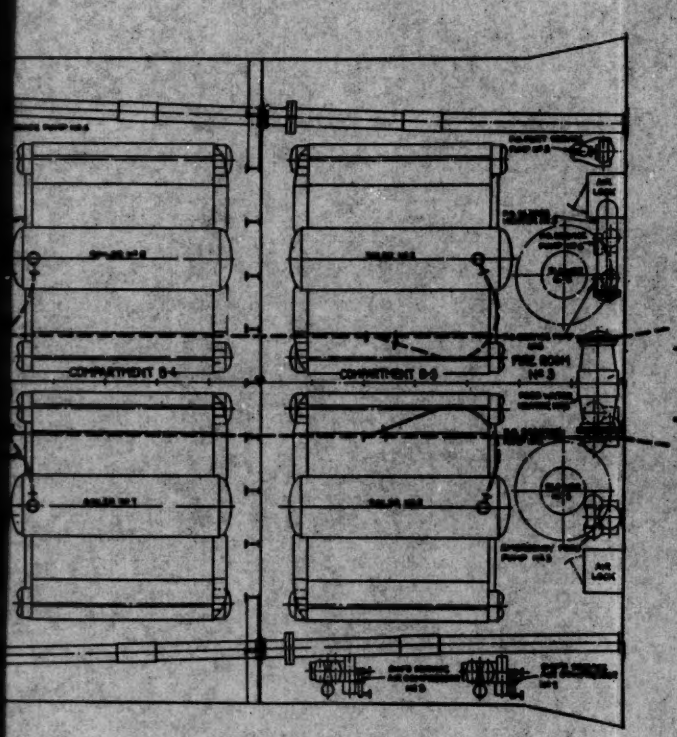
Averages		*Chester*	*Northampton*	*Houston*	*Salt Lake City*
10 knots	R.P.M.	92.24	90.61	94.57	90.69
	S.H.P.	1,964	2,126	2,021	1,374
	Speed	10.08	9.90	10.25	10.22
15 knots	R.P.M.	139.05	139.67	138.85	129.65
	S.H.P.	6,114	6,522	6,053	4,695
	Speed	15.24	15.2	14.98	14.36
17½ knots	R.P.M.	163.26	162.41	164.82	158.54
	S.H.P.	9,663	9,919	9,878	8,651
	Speed	17.72	17.51	17.76	17.47
20 knots	R.P.M.	189.48	182.58	186.52	188.70
	S.H.P.	15,184	13,828	14,298	14,954
	Speed	20.48	19.56	19.9	20.6

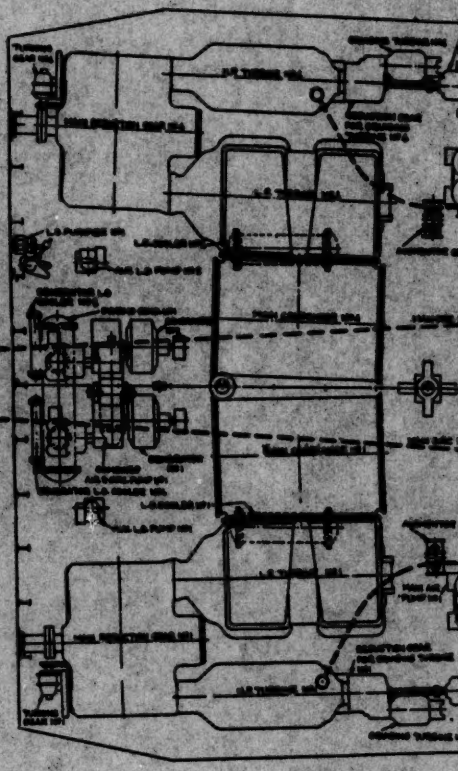


U.S.S. CHESTER (CL-27)
AFTER ENGINE AND BOILER ROOMS

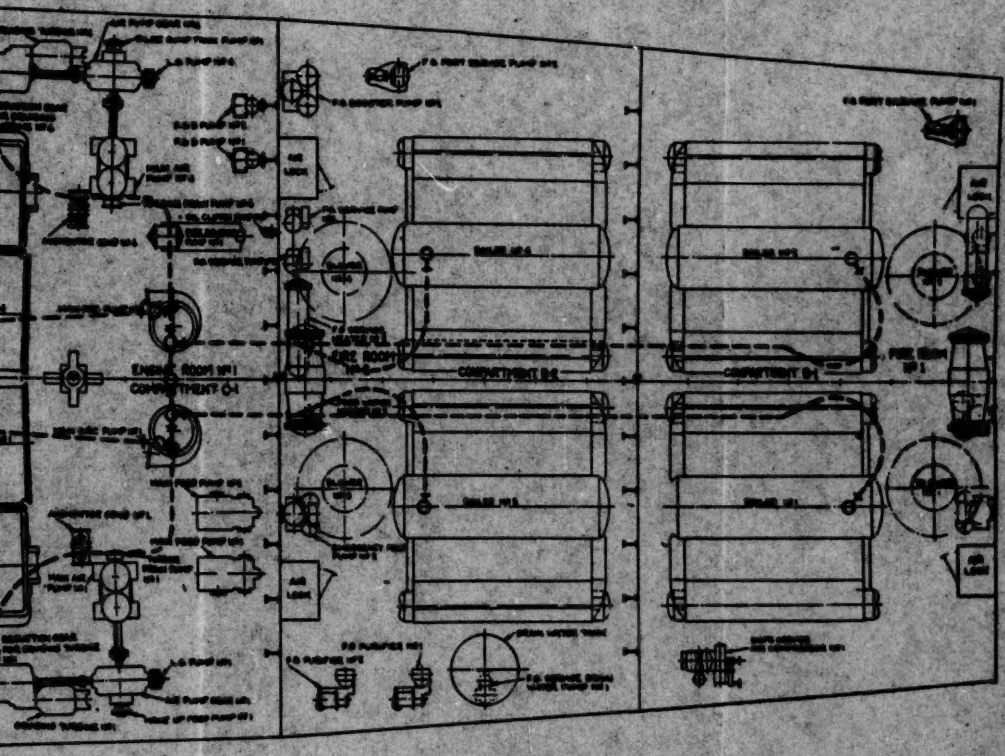








ER (CL-27)
BOILER ROOMS



U.S.S. CHESTER (CL.-27)

FORWARD ENGINE AND BOILER ROOMS

Averages		*Chester*	*Northampton*	*Houston*	*Salt Lake City*
22½ knots	R.P.M.	209.87	211.65	209.84	232.73
	S.H.P.	20,560	20,962	20,211	28,508
	Speed	22.57	22.68	22.40	24.79
25 knots	R.P.M.	236.41	236.6	233.2	236.94
	S.H.P.	29,057	28,421	27,699	29,964
	Speed	25.13	25.07	24.64	25.04
27½ knots	R.P.M.	260.96	264.87	261.25	268.01
	S.H.P.	39,403	41,772	39,073	44,659
	Speed	27.75	27.9	27.49	28.23
30 knots	R.P.M.	294.55	294.80	293.30	299.21
	S.H.P.	58,444	59,567	58,015	63,686
	Speed	30.05	29.94	29.92	30.20
31½ knots	R.P.M.	328.60	317.61	325.59	355.34
	S.H.P.	79,836	74,331	77,061	98,833
	Speed	31.57	31.10	31.34	32.22
Full speed	R.P.M.	380.59	374.38	380.75	372.06
	S.H.P.	109,062	111,461	110,814	107,746
	Speed	33.09	33.19	33.18	32.78
Astern	R.P.M.	259.83	248.56	250.84	255.76
	S.H.P.	33,271	32,175	30,698	26,076
	Speed	21.57	21.25	21.49	21.04

DATA TAKEN ON OTHER RUNS.

	Chester	*Northampton*	*Houston*	*Salt Lake City*
2 Hour 10 Knot Main Engine.				
Average R.P.M.	91.80	91.67	93.43	89.58
Average speed knots	10.09	10.00	10.14	10.07
Average S.H.P.	1,990	2,570	2,028	1,497
4 Hour 10 Knot Cruising Combination.				
Average R.P.M.	92.24	90.01	92.57	89.79
Average speed knots	10.13	9.82	10.06	10.09
Average S.H.P.	2,000	2,486	1,837	1,694
2 Hour 15 Knot Main Engines.				
Average R.P.M.	138.54	138.34	140.34	136.85
Average speed knots	15.11	15.01	15.13	15.15
Average S.H.P.	6,000	6,980	6,212	5,849
4 Hour 15 Knot Crusing Combination.				
Average R.P.M.	136.62	136.46	139.25	135.74
Average speed knots	14.90	14.81	15.04	15.03
Average S.H.P.	5,800	6,583	5,981	5,685
2 Hour 20 Knot Trial Main Engines.				
Average R.P.M.	186.72	185.43	187.01	183.53
Average speed knots	20.18	19.82	20.05	20.06
Average S.H.P.	14,500	15,203	14,439	13,678

	Chester	*Northampton*	*Houston*	*Salt Lake City*
4 Hour 20 Knot With Cruising Combination.				
Average R.P.M.	185.16	180.82	187.05	183.32
Average speed knots	20.02	19.37	20.05	20.01
Average S.H.P.	14,100	14,143	14,580	14,077
4 Hour, 29,560 S.H.P. Trial.				
Average R.P.M.	239.07	238.23	239.47	236.53
Average speed knots	25.40	25.26	25.31	25.07
Average S.H.P.	29,900	30,102	30,068	30,146
4 Hour 62,600 S.H.P. Trials.				
Average R.P.M.	300.34	299.88	303.99	296.61
Average speed knots	30.35	30.23	30.38	30.06
Average S.H.P.	61,800	63,399	63,804	63,006
4 Hour Full Power Trial.				
Average R.P.M.	378.92	371.42	375.17	370.07
Average speed knots	33.04	33.10	33.02	32.70
Average S.H.P.	108,100	110,314	108,231	108,687
2 Hour Maximum Power Trial.				
Average R.P.M.	380.42	373.36	379.52	372.33
Average speed knots	33.08	33.17	33.15	32.77
Average S.H.P.	109,000	111,302	109,849	109,657

The variation in power and speed between ships is in part due to slight differences in displacement and trim and to weather conditions encountered. Speed as given is taken from standardization curves.

Machinery in general performed very satisfactorily with the exception of main feed system, with which trouble was experienced on all vessels when operating under certain severe maneuvering conditions.

The maximum allowable bonus was earned by the contractors for all vessels both in fuel economy and in saving in weight.

All vessels exceeded their contract speeds and horsepower without undue forcing of boilers or machinery.

NOTES.

INDEX TO NOTES.

INDUSTRY DEDICATES A SHRINE TO GEORGE WESTINGHOUSE.
Electrical World, October 11, 1930.

THE DESTRUCTION OF THE "R 101".
The Engineer, October 10, 1930.

HIGH PRESSURE BOILERS FOR THE S. S. "KING GEORGE V".
Engineering, October, 3, 1930.

PRINCIPLES OF SELSYN EQUIPMENTS AND THEIR OPERATION.
General Electric Review, September, 1930.

MACHINERY INSTALLATIONS OF THE LINER "BREMEN" AND ITS PREDECESSORS.
Schiffbau, October 1 and 15, 1930.

INDUSTRY DEDICATES A SHRINE TO GEORGE WESTINGHOUSE.

By Earl Whitehorne

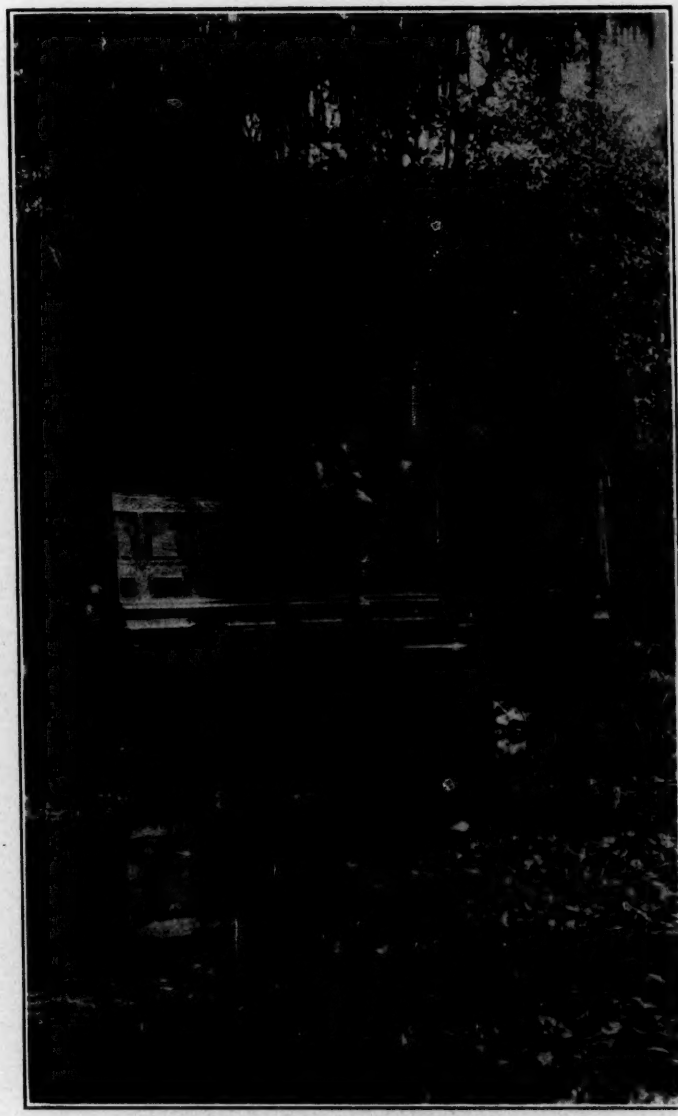
Contributing Editor "Electrical World."

George Westinghouse lived again in his industry this week, when, before a vast gathering of prominent Americans and the veterans of the Westinghouse organization, a monument was dedicated to his memory. It was an expression of the pride and veneration of more than 54,000 Westinghouse employees who contributed to a memorial fund to do honor to this great man. And the occasion in itself was a recognition and a tribute such as have seldom been given in the history of industrial progress.

It was a brilliant and impressive picture: More than 800 men and women, grouped at the entrance to a wooded ravine in beautiful Schenley Park in Pittsburgh. The clear warm sunshine of a perfect autumn afternoon. Before them, bordered on either side by the broad green of two weeping willow trees, a lily pool. Beyond it, against a background of rocks and the first color of the fall foliage, the memorial. It is a large central panel flanked by two semi-circular wings, all in gold-leafed bronze, with base and insets of black Norwegian granite. In a medallion on the central panel George Westinghouse leans over his drawing board. On either side stand life-size figures in full relief—a skilled mechanic and an engineer with his slide rule. Each of the flanking wings bears three panels depicting in low relief the major events in the career of Mr. Westinghouse. Then, before the monument, placed well back from the panels stands a bold figure of American youth, the work of Daniel Chester French. Just a boy it is, in knickers and sweater, his schoolbooks under his arm, who gazes with eager, reverent interest upon this inspiring testimonial to creative genius.

No one fortunate enough to participate in these ceremonies of dedication will soon forget the figure of this youth standing in silent revery before George Westinghouse. The boyish figure and the beautiful memorial background reflected in gleaming gold in the pool below, the throng of silent spectators, the chorus and band of Westinghouse employees, the ranks of veterans, many of them contemporary workers with George Westinghouse, and the host of distinguished guests, leaders in the electrical and railway industries, the financial and business world, massed about the speakers' platform made a vivid scene. It was impressive because this throng had come together to honor not only a great man but the ideals and qualities of greatness which that man personified. Through the characteristics of his life and work, therefore, this shrine of George Westinghouse becomes "a monument to the dignity of labor and the supreme glory of service." It is a symbol of the faith, courage and spirit of man that have built the industrial empire.

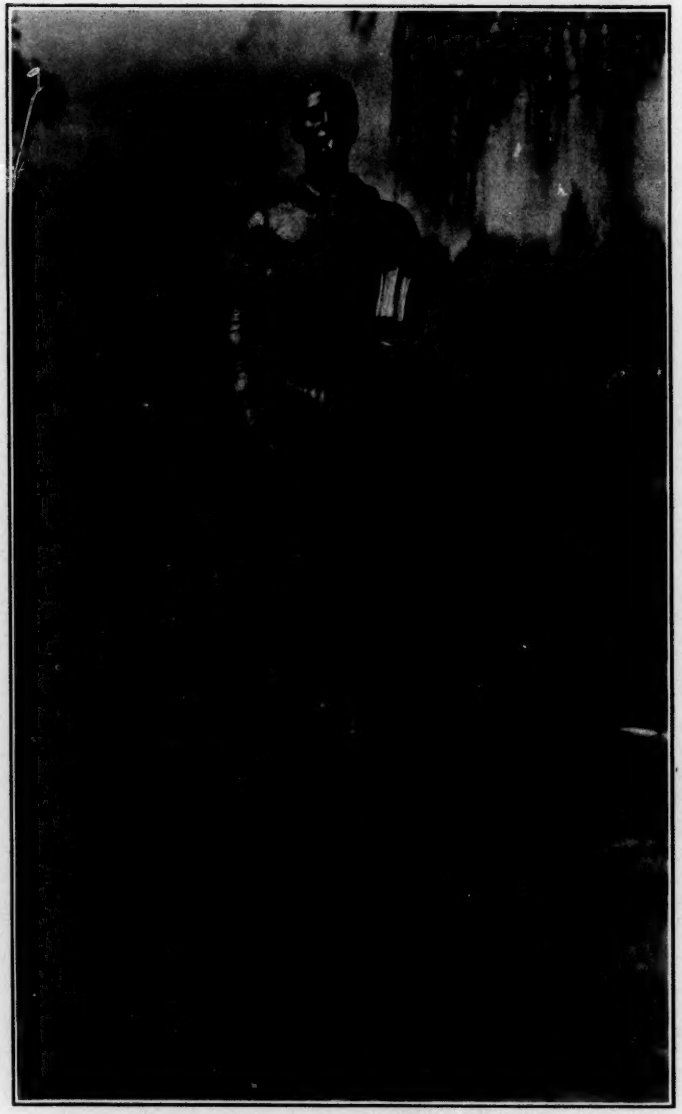
The story of the life of George Westinghouse was told in varied detail by eloquent speakers in the program of dedication and at the banquet of 700 guests of the Westinghouse organization which assembled at the William Penn Hotel in the evening. It is a romance of rich and versatile imagination, of unceasing initiative and unselfish toil in the pursuit of a broad vision of progress—a romance dramatic in its inception. A young man in his teens witnesses a disastrous train wreck in the old days of hand-braking and is deeply impressed. He becomes absorbed in the problem of promoting safety in railroading and finally conceives and produces the air brake. Against the dogged opposition of intrenched tradition and the criticism of the ubiquitous



GEORGE WESTINGHOUSE MEMORIAL.

ERECTED IN SCHENLEY PARK, PITTSBURGH, PA.

DANIEL CHESTER FRENCH, SCULPTOR—HENRY HORNBOSTEL, DESIGNER.



scoffer, the young man fights his way forward until he overcomes skepticism and establishes the principle of pneumatic control in transportation. He revolutionizes railway practice, first in America and then in Europe, by the sheer force of his indomitable effort and founds a great industry upon his invention. He then turns his attention to the possibilities of automatic signaling for railroads, establishes the principles in use today and organizes still another company to manufacture this safety device. In commenting at the banquet on these two contributions, James M. Beck voiced the opinion that Mr. Westinghouse had done more than any other one man to fulfill the prediction of George Washington that the progress of the American nation depended above all else upon the development of the economic bonds of contact and communication between the states. For it was through the control of trains by air brake and signaling that the expansion of railway transportation became possible.

The achievements of George Westinghouse in the field of electrical invention and manufacture were no less spectacular and constructive. He became the champion of the new principle of alternating current, believing that it would solve the perplexing problems involved in the transmission of power. He journeyed to England in 1885, purchased the transformer patents owned by Goulard and Gibbs and the next year organized another industry for the manufacture of electric lighting apparatus. The story is well known to all electrical men. He engaged Nikola Tesla, a young European scientist of great talent, to collaborate in the development of the alternating-current induction motor. He fought a crusade against the most bitter opposition and in 1892, by his personal efforts, obtained the contract to light the Chicago World's Fair and achieved a brilliant demonstration of the efficiency of his alternating-current system. Two years later he erected the world's first important hydro-electric system at Niagara Falls. He perfected the first alternating-current electric locomotive and was a pioneer in the early American development of the steam turbine—all major milestones in his progress. Meanwhile he pursued an inventive activity of astonishing proportions, embracing a complete automatic control telephone exchange system, an automatic coupler for air, electric and train connections, a system of piping and meters for natural gas, an air spring for automobiles and innumerable other patentable inventions of a surprising diversity that in many cases mirrored an uncanny vision of future possibilities.

And this in addition to the organization and administration of a group of industries whose operations embraced, in addition to America, Great Britain, France, Italy and Russia and involved the vicissitudes of repeated financial stress and reorganization—the price of continuous pioneering and expansion. He died in 1914, at the age of 68, rich in the record of achievement and in honors received at the hands of his countrymen, from rulers and from kings. He left behind him industries upon which 200,000 people were dependent for their wellbeing.

George Westinghouse was one of those great and unique figures who at rare intervals cross the pages of history. It is impossible to measure him in terms that apply to ordinary men. He patented over 400 inventions, but he was not primarily an inventor. He founded scores of successful companies, but he was in no sense a promoter. He made millions for himself and others, but he cared nothing for money for its own sake. He directed many vast engineering undertakings, but he cannot be considered merely an engineer. He was one of the most impressive figures, one of the most dynamic forces of his generation, and he applied his gifts and his ability for the benefit of humanity. He contributed greatly to the material develop-

ment of his country and the world at large and, because of the universality of his work and genius, belongs not only to America, but to the world.

James Francis Burke, in the dedicating address before the monument, stressed the abiding confidence of Westinghouse in the destiny of his country. He said:

"George Westinghouse's inventions and industries were not the only heritage he handed down to his successors. His faith in himself and in the future of America were outstanding features of his life. He transmitted that faith, and all the inspiration it imparts, to those who came after him.

"No more striking example of the value of men who in periods of depression and adversity keep their heads cool and their faces to the front can be found in American industrial history than his unceasing faith and fighting spirit during the most discouraging period of his financial and industrial career.

"The instrumentalities he created and the forces he set in motion enabled him, not only during his life, but in a greater measure after death, to triumph on the battlefields of peace, for there, after all, the most enduring of all man's victories are achieved. And in the light of the history that followed how strikingly was his faith and judgment vindicated!"

A. W. Robertson, chairman of the board of Westinghouse Electric & Manufacturing Company, speaking as toastmaster at the banquet, paid this tribute:

"George Westinghouse was like the rest of us in that he had two eyes with which to see, two ears with which to hear and the usual physical attributes that are common to all of us. In this sense he was an ordinary man—but here the 'ordinary' ceased and the 'extraordinary' began. His eyes were not limited to seeing the world of his day only, but he had the vision of a prophet and a seer—to look into the future.

"He was among the first of living men to visualize two generations ago the present age in which we live. He saw clearer than other men that the time would come when the wasted energy of the waterfall and the sleeping energy of the coal mine would be turned into the restless energy of the electric current, spreading everywhere throughout our country, to be the ready servant of the world. He had the energy of an army. He conceived and carried on huge projects. He rushed from one to another with unabated zeal. His was the faith that moves mountains. He had the courage that is equal to every emergency. His is an immortal name.

"He passed away a decade ago, and yet his name is better known than before his passing, and today we delight to honor him. No history of America or of the nineteenth century will be written without the name 'Westinghouse.' It is known in every language and in every land. So I consider it no idle statement to say that he belongs to that few hundred whom the world remembers and continues to honor.

"The bronze monument which we have dedicated this day to his memory in Schenley Park will endure as long as our present civilization lasts. We have erected it, not in the thought of making his name and achievements live, for they will endure regardless of monuments or bronze, but rather as a permanent expression of the love and respect which the men and women of Westinghouse have for him and his memory.

"His career was typically American. He was born without riches but in his liftime he gathered riches and honors of untold worth. He saw his dreams come true. The products of his laboratories and shops, children of his brain, are known in the far-flung corners of the earth.

"If by any chance tonight he looks down from the ramparts of Heaven upon this great gathering in his honor, may we express the wish that his

spirit has free access to the workshop of the universe, the research laboratory of the skies. If he accomplished so much in the space of one short lifetime, what marvelous radiant instruments may one visualize his restless brain creating in the eternal, changeless years of those who dwell on the other side of time! May he work there without being weary!"—"Electrical World," October 11, 1930.

THE DESTRUCTION OF THE *R 101*.

"His Majesty's Airship '*R 101*' is a huge, a courageous, and, it is certain, an expensive experiment; but, first and last and always an experiment." It is a year since we wrote these words. The melancholy task now befalls us to record that the experiment has failed. The airship lies a burnt-out wreck at Beauvais in France. Of the passengers and crew, numbering fifty-four, who left Cardington in her on Saturday evening for Karachi, all but six are dead. Excluding human error or failure, four possible broad sources of the disaster can be specified. There may have been a failure in the structure of the ship. There may have been a failure in her engine and fuel system. There may have been a failure in her navigational equipment, or there may have been no specific failure but a defect which might be described as aerodynamical unfitness, and which might have been discovered had her trials been prolonged. The accounts received from the survivors and from witnesses on the ground are incomplete and conflicting. In any event the Air Council is arranging to hold a public inquiry into the disaster. Even though we were fully informed as to its circumstances, it would, therefore, be undesirable at this instant to express any detailed opinion on the cause or combination of causes which produced it. Rather let us recall some of the salient facts in the technical history of the ship in order that we may prepare ourselves to follow with understanding the evidence and findings of the Court of Inquiry.

When Mr. MacDonald's Government in 1924 sanctioned a program of airship development, involving the construction of two experimental vessels, it had in view primarily the possible use of the airship as a means of improving Empire communications. Active steps were taken at an early date to interest India and the Dominions of Canada, South Africa, Australia, and New Zealand in a scheme to provide at first an experimental and later a regular airship service. With the exception of Canada, that scheme of necessity covered the operation of airships in tropical or semi-tropical climates. During the war, in 1917, a Zepplin airship, the *Z 57*, had flown from Bulgaria to Central Africa with supplies for the German troops operating in that region. The voyage marked the first occasion on which an airship had passed from northern to tropical latitudes, and in this respect it remains today the only one of its kind. It was a daring feat, justified only by the necessities of wartime. Its daring arose from the fact that the engines used petrol as fuel. It is impossible in an airship to prevent some leakage of petrol in the long lines of piping leading from the containers to the engine cars. The fuel tanks have necessarily to be distributed along the length of the hull, and cannot be concentrated near the already concentrated loads represented by the engines. Even in a cool climate the internal spaces of a petrol-driven airship soon become charged with fumes. In a hot climate the trouble would be magnified to a source of considerable danger. With these facts in view, it was decided that the two airships of the 1924 program should be driven by heavy-oil engines. Engines of that type were not then immediately available for aeronautical use. Work was begun on their de-

velopment, but various causes of delay arose, and it became apparent that if the airships were to be ready within a reasonable time heavy-oil engines could be provided for but one of them. The *R 101* was selected to receive the engines developed by Messrs. Beardmore. The *R 100*, equipped with Rolls-Royce aero-engines using petrol, was earmarked for trial on the Canadian route. The use of heavy oil engines on the *R 101* was in its inception not so much an experiment as a necessity for the service on which she was to be employed. The actual engines fitted to her were, however, purely experimental, and round them centers the main subsequent technical history of the vessel. In the first place, as delivered they were several tons heavier in the aggregate than the estimated weight, and consequential increases in weight had to be made in portions of the ship's structure. Secondly, it was found on trial that they were subject to serious torsional oscillation in their crank shafts. Improvements were subsequently effected in this respect, but the trouble persisted to such an extent that an alteration had to be made in the designers' intentions. It had been proposed to run all five of the engines in a constant direction, and to drive the vessel astern by the use of propellers with reversible blades. The torsional oscillation was found to throw excessive stress on the blades at their union with the boss, and the idea of making them reversible had to be given up. Fixed-blade propellers were used instead, one of the five engines being equipped with a reversed pitch propeller, and being used solely for driving the vessel astern. In this condition the vessel was launched, and put through her trials. Following her appearance at the Royal Air Force display last June, she was returned to her shed at Cardington. She had by all accounts behaved satisfactorily, but the expedients which had been adopted to meet the characteristics of the engines had resulted in increasing her deadweight, and therefore decreasing her carrying capacity, and in reducing her forward driving power and therefore lowering her speed. It was determined to remedy both these defects. First as regards the deficiency of power, it was arranged to install two reversible engines in the forward cars. Engines of that pattern had in the meantime been developed, and it was therefore possible to equip the vessel with five engines, all of which would be available for forward driving, and two of which could be used for reverse running. Secondly, as regards the deficiency of lift, three principal directions were followed. The gas bags were let out a few inches all around, the original clearance between them and the framework having, so it was said, been found to have been fully liberal. Originally a servo-motor system had been provided to facilitate the operation of the rudders and elevators. It was found quite possible to operate these organs by hand, and consequently it was decided to save weight by discarding the servo-motor equipment. Then as the major source of obtaining increased lift, the vessel was cut in two and an additional bay 50 feet in length, containing a new gas bag, was built into her structure. On October 1st she emerged from her shed in her reconstructed form, and proceeded on a trial flight of sixteen hours' duration. As enlarged, she had a length of 777 feet, a gas capacity of 5½ million cubic feet, and a gross lift of 169 tons. Several minor alterations were also effected in her design, the most notable of which was that she was fitted with a system, previously tested in the *R 100*, for catching the rain water which might fall on her hull and using it as ballast to compensate for the consumption of stores and fuel. The vessel was experimental, not only in her power equipment, but in the design of her structure. What that structure was, and how it was fabricated at the works of Boulton and Paul, we have previously detailed. See "The Engineer," August 31, November 30, and December 7, 1928. It is sufficient to say that it was composed largely of stainless and high tensile steel, instead of the usual duralumin, and was arranged in an entirely novel

manner. It should be added that sections of the structure were proved by tests, and that they fully came up to expectations as regarded strength and lightness. No modifications, so far as we know, were at any time made in the structure for purely structural reasons. Unlike the engines, the structure, although experimental, contributes nothing to the technical history of the vessel. Indeed, it may be said that the structural formation adopted rendered possible, or at any rate easy, the lengthening of the hull, which was undertaken to compensate for the shortcomings of the engines.

On August 23, 1921, the airship *R 38* broke in the air over Hull with the loss of forty-four lives. There were many people, who, shocked by that disaster, felt strongly that it ought to be made to mark the end of airship building, so far as this country was concerned. Before the year was out, however, the Airship Stressing Panel had been appointed to study the stresses in the hulls of airships in light of the facts revealed by the disaster. Those stresses had previously been guessed at, rather than calculated. It had been discovered that a serious omission of knowledge had existed. The *R 38*, it was revealed by scientific investigation, broke her back because her framework, while strong enough to support the normal aerodynamic forces, was not strong enough to take the unsuspectedly high dynamic forces thrown upon it when the vessel endeavored to execute a sharp turn at high speed. Here was a good reason, so it was urged, for giving the airship another chance. The argument was difficult to resist, and eventually it won the day in 1924, when the Government consented to the program of development involving the construction of *R 100* and *R 101*. On September 3, 1925, the American airship *Shenandoah*, a helium-filled vessel, broke in two in the air, with the loss of fourteen lives out of a crew of forty-two. Once again science stepped in. It was discovered that vertical currents of air of unsuspected intensity sometimes accompanied a thunderstorm. In such a current the airship had been caught, and violently tossed up several times. Her structure had not been designed to withstand any such motion, and perforce failed. The argument carried conviction and was not allowed to stop the airship activities of the country. The United States is now, in fact, building two airships, each of 6½ million cubic feet capacity. Other airship disasters have taught other lessons, but the two which we have mentioned are outstanding in the repercussion which they had on the design of *R 101*. That vessel was constructed in full knowledge of the causes which had wrecked the *R 38* and the *Shenandoah*. Her *early* trials proved that she was capable of executing a sharp turn at high speed. She had never encountered a serious vertical current, but there is not much possibility for doubt that, in at least her *original* condition, the allowance made for such an event was ample. In support of that statement, it may be said that the *R 100*, designed to the same general strength specification, encountered and survived unhurt a severe vertical current towards the end of her outward voyage to Canada. Yet in spite of all the care, knowledge, skill, experience, and foresight which went to the making of the *R 101*, the vessel has failed as suddenly, and, for the moment, as inexplicably, as any of her forerunners. Is it possible that once again scientific inquiry, wise as always after the event, will establish some hitherto unsuspected factor as the cause? Is it possible that, if it does, men will be found bold enough to urge once again that the airship should be given another chance? With a heavy heart, we are constrained to answer both questions with a "Yes." It would be against our convictions, but not against our wishes, if we answered that the disaster to *R 101* will assuredly mark the end of the airship. She was an experiment. The experiment has failed, but how near was it to success? We do not know, and may never know. It is, however, certain that when the shock of Sunday's catastrophe has become dulled by time, arguments will be advanced justifying, or at-

tempting to justify, the continuation of the experiment. Some trifling cause, some minor error of judgment or execution, not some omission of a major scientific fact, may be the explanation of the disaster. If such proves to be the case, is it conceivable that the experiment will be permanently abandoned? We cannot believe it. We cannot hope that science, taking it and all the lamentable previous occurrences of its kind, will turn from finding an excuse for them, and instead use them to prove that the major and minor factors affecting airship construction and operation are so numerous, so complicated, and so much influenced by chance, as to leave the ambitions of their advocates outside the reach of practical realization.—"The Engineer," October 10, 1930.

HIGH-PRESSURE BOILERS FOR THE S.S. *KING GEORGE V.*

It is not often that large-scale experiments in marine steam-raising plant can be made in a practical form over long periods, so that it is interesting to record an instance of the kind. This lies in a change in the type of boiler fitted to the Clyde passenger turbine steamer, *King George V.* It will be recalled that this steamer, which has the highest working boiler pressure of any vessel afloat, was built not only to demonstrate the higher efficiencies resulting from the employment of higher pressures than have hitherto been customary at sea, but, to use Sir Charles Parson's own words,* "to explore the application of high temperatures and water-tube boilers to marine work, and also to ascertain what practical difficulties, if any, had to be overcome in this connection." This exploration is now carried into another stage by the installation of Babcock and Wilcox water-tube boilers in place of those originally fitted, an amicable agreement having been arrived at between the two boiler-making firms concerned and the managing company of the steamer, these firms being Messrs. Babcock and Wilcox, Limited, Farringdon-street, London, E.C.4, Messrs. Yarrow and Company (1922), Limited, Scotstoun, Glasgow, and Messrs. John Williamson and Company, 308, Clyde-street, Glasgow, C.1, respectively.

The first boiler installation, which consisted of two five-drum Express-type boilers, was fully described and illustrated in our account of the steamer when first put into service, *i.e.*, in "Engineering," vol. cxxii, page 321 (1926), and we now give in Figures 1 to 3, some corresponding views of the new arrangement. It will be seen from Figure 1, that, as before, the boilers are facing one another, and the two funnels are retained. The coal bunkers have not been touched, the width of the stokehold being the same as before, but the casing underneath the funnels has been somewhat raised, and there are some changes in the draught arrangements. To ensure that the results should not be affected by any abnormality of design, the boilers have been made of the usual Babcock and Wilcox construction with transverse drums, as is customary with marine practice. Each boiler is 18 sections wide, each section comprising a single bottom tube, 4 inches in external diameter and 24 tubes, arranged in pairs horizontally, above it. These tubes are 1 13/16 inches in external diameter, and all the tubes are 9 feet 3 inches long overall. The return tubes, two per section, are 4 inches in external diameter. The steam drum is 3 feet in internal diameter by 14 feet long overall, and access is gained to it by a 15-inch by 11-inch manhole at each end. It is made of seamless forged steel, of adequate thickness to give the necessary factor of safety in the shell, but with thicker areas in way of the mountings, return tubes and downtake nipples. These areas represent the original thickness, the shell having been reduced by machining in order to keep down the weight.

* See "Engineering," vol. cxxiii, page 87 (1929).

The tube nest is divided into three vertical passes by means of two vertical plate baffles and one of refractory material lying on the top of the two bottom rows of small tubes. The superheater is of the U-tube type, and lies horizontally in a pocket above the return tubes of the boiler. It has circular headers of the shape indicated in Figure 1. An air heater, consisting of a nest of straight tubes, is situated above the superheater. It is supported independently of the boiler, while the weight of that portion of the uptake above it, and of the funnel, is also taken independently. The heated air passes down a tapering trunk to the back of the boiler, over which it is led, in the passage visible in Figure 1, to the underside of the firebars. The grate is 6 feet 10½ inches long and extends the full width of the boiler, being fired through three doors hinged at the top, opening inwards and operated by levers. The ashpits are closed with butterfly doors. The boilers are operated on the plenum system, that is, the stokehold is closed, and a low air pressure is maintained in it by fans so as to ensure a proper flow of secondary air through the fire doors. This involves entering the stokehold through air locks, but, as the pressure is only about ⅞ inch of water, there is no real inconvenience. It is claimed, with reason, that an economy is probably effected, inasmuch as the radiated heat is largely carried back into the furnaces in the primary air entering the heaters and passing under the firebars. The heater air supply is regulated by the hinged dampers seen in front of them. In addition, as the air pressure is external to the boilers, no leakage through the casings can enter the stokehold. The air supply to the fans is taken through a coaming covered with a grid and situated on the main deck against the boiler casing in the starboard alleyway, *i.e.*, immediately over the fan room, shown in Figure 3.

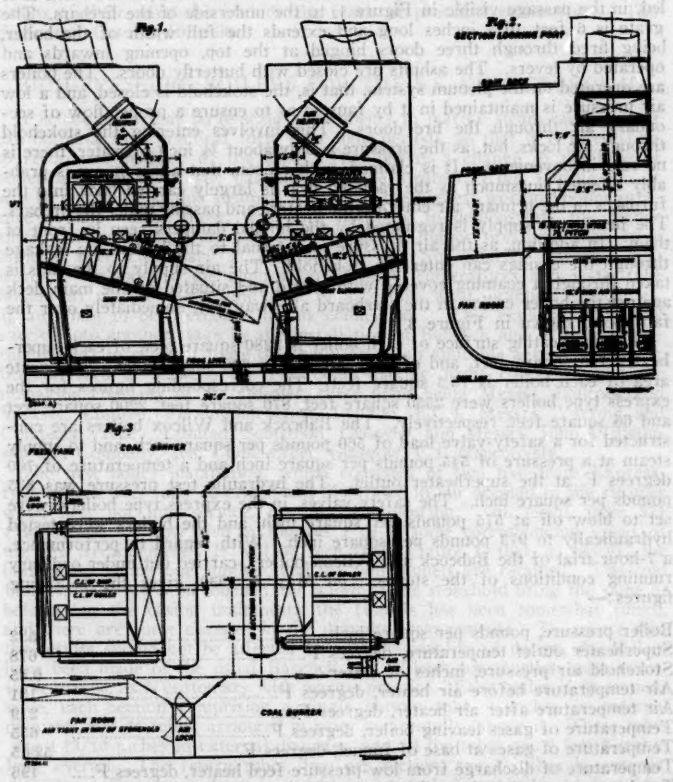
The tube heating surface of each boiler is 2280 square feet, of each superheater 780 square feet, and of each air heater 800 square feet. The grate area of each boiler is 75.5 square feet. The corresponding figures for the express type boilers were 2550 square feet, 870 square feet, 2200 square feet and 66 square feet, respectively. The Babcock and Wilcox boilers are constructed for a safety-valve load of 560 pounds per square inch, and to supply steam at a pressure of 545 pounds per square inch and a temperature of 700 degrees F. at the superheater outlet. The hydraulic test pressure was 875 pounds per square inch. The safety valves, in the express type boilers, were set to blow off at 575 pounds per square inch, and the boilers were tested hydraulically to 913 pounds per square inch. With regard to performance, a 7-hour trial of the Babcock and Wilcox boilers, carried out under ordinary running conditions of the steamer, on May 8, 1930, gives the following figures:—

Boiler pressure, pounds per square inch	532
Superheater outlet temperature, degrees F	678
Stokehold air pressure, inches of water	0.85
Air temperature before air heater, degrees F	131
Air temperature after air heater, degrees F	219
Temperature of gases leaving boiler, degrees F	655
Temperature of gases at base of funnel, degrees F	524.5
Temperature of discharge from low-pressure feed heater, degrees F	196
Total water evaporated per hour, pounds	33,620
Coal used per hour, pounds	3578
Calorific value of coal ("Polmaise") B.T.U. per pound	13,990
Thermal efficiency of boiler, per cent	79.6

It would serve no useful purpose to quote the corresponding figures for the Express type boilers, but it is interesting to note that the efficiency above recorded is just slightly better than that given for the first installation for

practically the same rate of evaporation. This latter figure will be found in Sir Charles Parson's paper on "Progress in Economy of Turbine Machinery on Land and Sea," reproduced in "Engineering," vol. cxxiii, page 87 (1927). It might also be noted that the area of the air heater in the Babcock and Wilcox installation is considerably less than that of the Express type boilers.

The boilers now described have been working for two summer seasons on the Clyde tourist and passenger traffic, and examination made at the end of the 1929 season showed that they had stood up to the trying conditions of the



service, which involve frequent rapid starting and stopping, in a very satisfactory manner, the only deterioration visible being some wear and slight distortion on some of the baffles, tube supports, and casing plates. During the time the vessel was laid up for the winter, some changes were made in these parts to secure greater durability, and one or two other alterations were made. These comprised the fitting of an additional gauge glass on the center of the steam drums, to facilitate correct reading of the water level

should the vessel have a list, an improvement in the air supply at the back part of the furnace, and the simplification of the means of controlling the saturated-steam supply from the de-superheater. Periodical internal and external examinations carried out during 1930 have shown the condition of the boiler to be thoroughly satisfactory.

The working of the boilers was demonstrated on Thursday, September 25, to a large number of marine superintendents, consulting engineers and others interested, on a special trip to Loch Fyne. The conditions in the stokehold, in spite of the high superheat temperature, were excellent, all working spaces being agreeably cool and free from the down draughts which often obtain when ventilators are used. The operation of firing, due to the draught system used, was not at all trying, though hot fires were maintained. These conditions were reflected in those on the main and promenade decks. In steamers of this type, where most of the passengers' time is spent on deck, there are often complaints of inconvenience, and sometimes of damage, from hot boiler-room casings, but we found no reason to apprehend anything of the kind in this case. The engine room also was not unduly hot, an improvement having been effected, we understand, by a change in the gland seals of the high-pressure turbine.

In the course of a speech delivered at the luncheon on board on the occasion of the demonstration trip, Sir Charles A. Parsons, in stating that the results had justified the experiments, said it would appear that engineers were beginning to think that the Scotch boiler would eventually be superseded by the water-tube boiler. This vessel must be considered as a stepping stone to the use of higher pressures in larger ships, and, if a pressure of 1000 pounds per square inch was indicated as practicable, then the efficiency of steam-driven machinery would be equal to that of the Diesel engine and the drain of capital to pay for oil could be stopped with great benefit to the native product, coal. He hoped that, in a few years, all large ships would be driven by steam at a pressure of 1000 pounds per square inch.

Vice-Admiral Sir Robert Dixon, speaking of the work of Messrs. Babcock and Wilcox in the matter of high pressures and temperatures, referred to the experimental boiler for a working pressure of 1400 pounds per square inch and a temperature of between 800 and 900 degrees F., which the party had inspected at Renfrew earlier in the day, and stated that it would be possible even now to adopt a pressure of 700 pounds per square inch and a temperature of 750 degrees F. in the Babcock and Wilcox boiler for marine work.—"Engineering," October 3, 1930.

PRINCIPLES OF SELSYN EQUIPMENTS AND THEIR OPERATION.

PRINCIPLES INVOLVED—SELECTIVE AND MULTIPLE OPERATION—THE DIFFERENTIAL SYSTEM—MECHANICAL CONSTRUCTION—APPLICATIONS.

By L. F. HOLDER,

AERONAUTICS AND MARINE ENGINEERING DEPARTMENT,
GENERAL ELECTRIC COMPANY.

GENERAL THEORY AND DESCRIPTION.

A Selsyn system includes small self-synchronous instruments used as an electrical means of transmitting accurate angular motion between two or more remote devices which cannot conveniently be mechanically interconnected. The accuracy of the motion transmitted varies with the amount of torque required by the load, the size of Selsyn instruments used, and the acceleration and operating speed of the system.

A Selsyn device is mechanically a miniature bipolar rotating-field three-phase alternator. The rotor is wound with a single-phase concentrated winding; the stator, with a three-circuit distributed Y-connected winding. The shape of the rotor and stator punchings is shown in Figure 1. Electrically, in normal operation, the Selsyn device acts as a transformer and voltages and currents existing in the instrument are all single-phase. The rotor winding is excited from an alternating voltage source. By transformer action, voltages are induced in the three elements of the stator winding, the magnitude depending upon the angular position of the rotor.

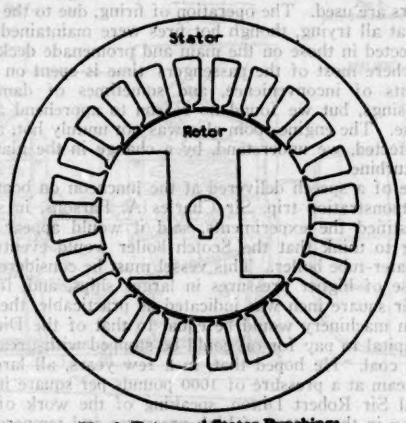


Fig. 1. Rotor and Stator Punchings

The simplest of the Selsyn systems consists of two instruments connected as illustrated in Figure 2. The two rotors are excited from a common source and the stator leads are interconnected S1 to S1, S2 to S2, and S3 to S3, respectively. Assuming both rotors free to turn, they will take such a position that the voltages induced in the two stators are of balanced magnitude and displacement. Under this condition of stator-voltage balance between the two instruments there is no circulating current in the stator windings. Now, if the rotor of one Selsyn device is displaced by a certain angle and the rotor of the other is held in its original position, the stator-voltage balance is altered and a circulating current will flow in the windings. This circulating current reacting on the excitation flux provides a torque tending to turn the rotors of the machines to a position where the induced stator-voltages are

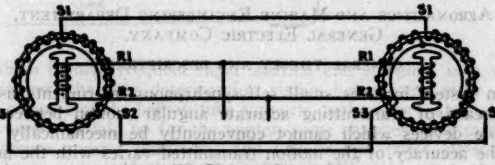


Fig. 2. The Simplest Selsyn System

again equal and opposite. Thus, with both rotors unrestrained, any motion given to the rotor of one Selsyn device will be transmitted to and duplicated by the rotor of the second, and we thus have a system of electrically transmitting mechanical motion.

It can be shown by mathematical analysis that the electromagnetic coupling between the two rotors is such that a load placed on the rotor of the receiving Selsyn device will cause it to lag behind the transmitting device by an angle which is dependent in size on the magnitude of the load. Figure 3 illustrates the torque, excitation current, and induced stator current for various angles of displacement. It will be noted that at 180 degrees displacement the circulating stator current and the excitation current are at a maximum although the torque is of zero value. This point is very unstable and Selsyn devices of their own accord will never operate with this displacement, but will always synchronize when their rotors are excited. This is one of the superior characteristics of the Selsyn system over other schemes of electrical transmission of mechanical motion; the system is self-synchronous, that is, at the resumption of power after an interruption the Selsyn system automatically resets itself.

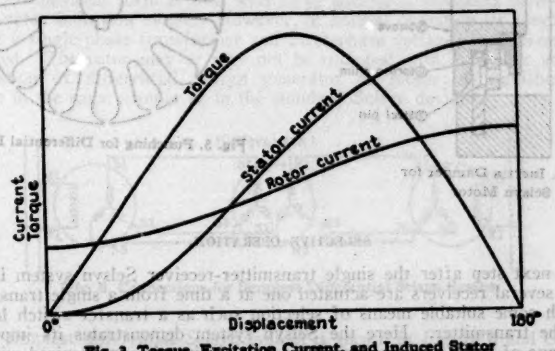


Fig. 3. Torque, Excitation Current, and Induced Stator Current for Various Angles of Displacement

The Selsyn instrument draws but little power from the line and is designed for continuous excitation. The electrical design is based on the allowable temperature rise in the instrument. Since the excitation current and the circulating stator current increase with the magnitude of displacement, a Selsyn system must never have a displacement angle exceeding 20 degrees for more than a short period; otherwise, excessive heating and possible burnout of the windings may occur. Thus, in applying Selsyn equipments to a particular duty the torque requirements should be carefully studied and equipment of ample size should be selected.

It has been the custom to call the Selsyn instrument which is used as the transmitter, a Selsyn generator, and the remote or receiving instrument, a Selsyn motor. The one does not generate electrical power nor does the other transform it into mechanical motion. The Selsyn system merely transmits mechanical motion and, neglecting friction losses, the mechanical power output of the receiver is exactly equivalent to the mechanical power input to the transmitters.

Electrically, the Selsyn generator and motor are identical. Physically, they differ in that the motor has mounted on its rotor shaft an oscillation damper. This damper consists of a lead ring mounted by friction plates on a sleeve which is fastened to the shaft. The lead ring has a large amount of inertia and can be rotated on the sleeve only with considerable difficulty. Any violent oscillation of the rotor rotates the sleeve, a motion which the lead ring cannot follow immediately because of its inertia. Thus, there is a large dampening effect on the oscillating rotor and its movement is quickly stopped. This inertia damper is illustrated in Figure 4.

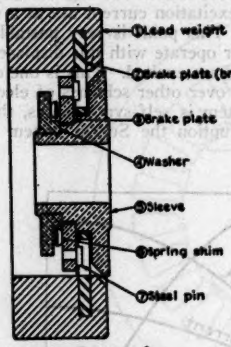


Fig. 4. Inertia Damper for Selsyn Motor

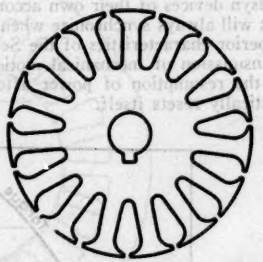


Fig. 5. Punching for Differential Device

SELECTIVE OPERATION.

The next step after the single transmitter-receiver Selsyn system is that where several receivers are actuated one at a time from a single transmitter through some suitable means of selection such as a transfer switch located near the transmitter. Here the Selsyn system demonstrates its important advantage of being self-synchronous. Any receiver is synchronized with the transmitter and can be operated at will, without the necessity of presetting, by a mere setting of the selector switch.

MULTIPLE OPERATION.

The Selsyn system may also be used for multiple operation; that is, several Selsyn receivers may be operated by a single Selsyn transmitter. The operation is in general the same as in the single transmitter and receiver system. There are, however, some additional inherent peculiarities that should be noted. In the system consisting of a single transmitter and receiver each of the same physical size, the torque-angle characteristic is considered as standard for that size of Selsyn frame. Now, if two receivers are operated from a single transmitter, all of the same frame size, the torque (T_R) developed by each receiver at a given angle of displacement will be only ⅔ of the standard value of torque (T_T). If there are three receivers, the torque of each is still further reduced to ½ T_T, and for any number of receivers $T_R = T_T \times \frac{2}{N+1}$ where N is the number of receivers. For

this reason it is standard practice in multiple Selsyn systems to use as the transmitter a device of larger frame size than the receivers, and of such characteristics that each receiver will develop its standard torque. Another peculiarity of the multiple Selsyn system where the friction load on the separate Selsyn receivers may vary is that the lightly loaded Selsyn receivers assist the transmitters to keep the more heavily loaded ones in synchronism, and much greater than standard torque may be developed by the more heavily loaded instruments. This is accompanied, however, by an increase in magnitude of the general system displacement angle.

SERIES OPERATION—THE DIFFERENTIAL SELSYN.

Series and series-multiple types of Selsyn systems are also used. These involve the use of the differential Selsyn device, the operating characteristics of which must be considered to get a correct understanding of its use.

The differential Selsyn device is identical in construction with the standard Selsyn instrument except that its rotor punching differs (see Figure 5) and the rotor has a distributed three-circuit Y-connected winding. Three collector rings instead of two are also necessary to bring out the rotor winding leads. Its physical form is thus seen to be that of a miniature three-phase wound-rotor induction motor. However, in normal operation its function is that of a single-phase transformer and three-phase voltages and currents do not exist. The rotor may or may not be equipped with a damper and the designation of differential Selsyn generator or motor distinguishes this feature in the same manner as in the standard Selsyn devices.

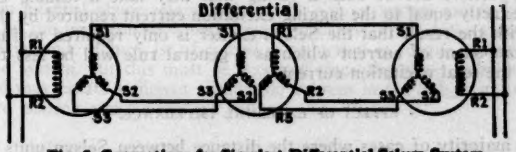


Fig. 6. Connections for Simplest Differential Selsyn System

The simplest differential Selsyn system is shown in Figure 6 and consists of three units—two standard and one differential. The stator winding of the differential is connected phase for phase to the stator of one standard Selsyn unit; the rotor winding is connected phase for phase to the stator of the other standard Selsyn unit. Since the differential device does not have a connection directly to the source of excitation, its exciting current must be furnished through one or both of the standard Selsyn devices to which it is connected. The general practice is to supply this excitation on the stator winding only. It is obvious that, since the standard Selsyn device connected to the differential stator supplies this excitation current, it must be designed to carry this extra load without overheating. A special design of standard apparatus is used, commonly designated "Selsyn Exciter," which may function in the system either as a transmitter or receiver, and sometimes in more complex systems merely floats on the system as an excitation supply to the differential.

The operation of the differential Selsyn device is very similar to that of the standard one. The voltage distribution in its stator winding is the same as in the stator winding of the Selsyn exciter. The distribution of the exciting flux is likewise the same. Also, the impressed voltage distribution in its rotor winding is the same as in the stator of the standard Selsyn unit to

which it is connected. The induced rotor voltage distribution is, of course, determined by the distribution of the exciting flux. The point of equilibrium is such that the impressed and induced voltages in the rotor winding of the differential Selsyn unit are equal and opposite. Under this condition there is minimum current flow. Disturbance of this condition will set up a circulating current which will react on the excitation flux, producing a torque tending to restore the equilibrium condition. It is apparent that the disturbance can be set up by moving the rotor of any of the three Selsyn devices. If any one of the three is fixed in position and a second one displaced a certain angle, the third, being free to rotate, will turn through the same angle. The direction of rotation can be reversed at will by reversing any pair of leads on either the rotor or stator windings, respectively, of the differential Selsyn units. If any two of the devices are rotated simultaneously, the third will rotate through an angle equal to the algebraic sum of their movements, the algebraic sign being dependent not only upon the physical direction of rotation of the rotors but also upon the phase rotation of the windings.

Thus we have a means of controlling a Selsyn receiver from two separate sources at will and, if the angle of rotation of the transmitters is limited, a self-synchronous means. A Selsyn system of this kind can be indefinitely extended in a wide variety of combinations of transmitters and receivers. Successful systems are in present use where three differential Selsyn units in series are linked between the primary transmitter and receivers. In these more complex systems, it is customary to relieve the Selsyn exciters from the necessity of furnishing the total exciting current. This is done by connecting balanced static capacitors across the stator leads of the differential. These capacitors are of such a rating that they take a leading excitation current exactly equal to the lagging excitation current required by the differential, with the result that the Selsyn exciter is only required to furnish the power component of current which as a general rule will be less than one-tenth of the total excitation current.

EFFECT OF EXTERNAL IMPEDANCE.

In the majority of cases where the distance between Selsyn units is short, the impedance of the connections can be neglected. Frequently, however, long connecting lines are necessary and, when the resistance of these lines becomes comparable with the resistance of the Selsyn stator winding, both the magnitude and phase angle of the stator current are affected. This change reduces the torque output which can be obtained from the Selsyn instrument. For this reason, in long Selsyn systems the resistance of the interconnections is made as low as practical and Selsyn units are used with stator windings of a high value of impedance. The torque-resistance characteristic and the effect of increasing the stator impedance are illustrated in Figure 7. For all except very special installations this method will suffice, although it is possible to obtain still better results by employing still higher voltages on the stator interconnections by the use of special transformers between the lines and Selsyn units.

As has been mentioned previously, all Selsyn units in a Selsyn system must be excited from the same alternating voltage source and, in case of multiphase supply, from the same phase of the supply. In some instances it is necessary to run lines to the remote Selsyn unit to supply this excitation. If these lines are long, their resistance will be large and there will be not only a considerable voltage drop in them but also an excitation current phase displacement in the remote Selsyn unit. Since the torque output is proportional to the square of the excitation voltage, and change in internal phase relationships in the system creates dangerous circulating currents, means

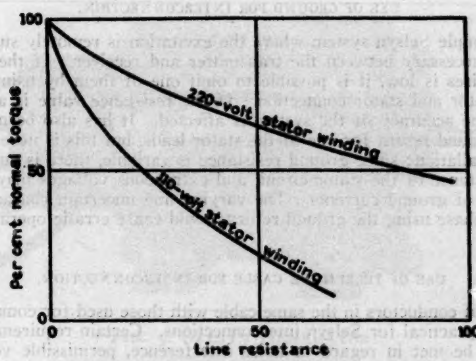


Fig. 7. Torque-resistance Characteristic and Effect of Increasing Stator Impedance

should be provided to remedy the situation when excitation must be remotely supplied. The standard means of raising the terminal voltage is to insert a small auto transformer in the line to compensate for the IR drop. The change in internal phase relationship in the Selsyn system can be taken care of only by the insertion of a balancing impedance in the rotor circuit of the near Selsyn unit. There is, of course, a reduction in torque output available from the system, but this must be accepted since the phase unbalance if not remedied may create sufficient circulating current in the Selsyn units to cause excessive heating and consequent failure. See Figure 8.

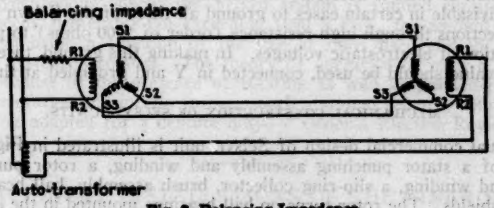


Fig. 8. Balancing Impedance

VOLTAGE RATINGS.

Standard commercial Selsyn units are wound to be excited from 110 volts, 60 cycles, but special forms are wound for other voltage and frequency ratings. The standard commercial stator voltage rating is 55 volts, but higher-impedance stator windings are commonly wound for 110 and 220 volts. In any Selsyn system, of course, the respective rotor and stator windings of all Selsyn units in the system must have the same electrical characteristics; that is, a unit with 55-volt stator winding cannot be operated with another having a 220-volt stator winding.

USE OF GROUND FOR INTERCONNECTION.

In the simple Selsyn system where the excitation is remotely supplied, five wires are necessary between the transmitter and receiver. If the resistance of these wires is low, it is possible to omit one of them by using one as a common rotor and stator connection. If the resistance value is appreciable, however, the accuracy of the system is affected. It has also been suggested to use a ground return for one of the stator leads, but this is never advisable on any installation; since ground resistance is variable, there is an unbalance of the resistance of the stator circuit and extraneous voltages may be picked up because of ground currents. The varying and uncertain characteristic of the stator phase using the ground return would cause erratic operation of the system.

USE OF TELEPHONE CABLE FOR INTERCONNECTION.

The use of conductors in the same cable with those used for communication is entitrely practical for Selsyn interconnections. Certain requirements, however, must be met in regard to noise interference, permissible voltage and currents, and necessary insulation. Filtering equipment is generally required between the source of excitation and the Selsyn apparatus. This must be of such design that there will be no appreciable rise in terminal voltage of the filter unit in case of open circuit and that there will be a reduction of at least 90 per cent in all harmonics above 300 cycles in going through the filter. Ground connection on the circuit electrically connected to the telephone conductors is not permitted and insulating transformers must generally be used to prevent the possibility of high voltages (approximately 3000 volts) being placed on the telephone wires. Operating voltages and currents must not exceed 120 volts and 350 milliamperes, respectively.

POWER LINE INTERFERENCE.

Where Selsyn unit interconnections closely parallel power lines, proper transposition of lines is necessary to eliminate inductive interference. It may also be advisable in certain cases to ground all three of the Selsyn unit stator interconnections through high resistance (order of 1000 ohms) to prevent the accumulation of electrostatic voltages. In making this ground, three resistors of equal value should be used, connected in Y and grounded at the Y point.

MECHANICAL CONSTRUCTION OF SELSYN UNITS.

The usual commercial design of Selsyn unit is illustrated in Figure 9 and consists of a stator punching assembly and winding, a rotor punching assembly and winding, a slip-ring collector, brush assembly, ball bearings, and cast end shields. The rotor turns on ball bearings mounted in the end shields and the stator and end shields are firmly clamped by three bolts in a rabbeted fit. By this means accurate centering of the rotor and stator punching assemblies is obtained. The commercial design is ordinarily foot-mounted, as illustrated, but is also built for flange and trunnion mounting. The collector in most of the later designs is external to the collector end shield and is covered with an easily removable inspection cap. The collector rings and brushes are both made of special materials to meet the unusual requirements of a low-friction non-corrosive unabrasive perfect electrical contact. Leads of both rotor and stator windings are brought out and after testing are tagged with the correct phase rotation. The Selsyn instrument is exceptionally compact and strong, and must pass rigorous noise and accuracy tests.

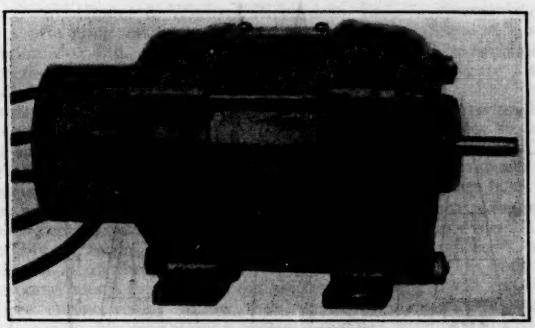


FIG. 9. SELSYN GENERATOR.

SPECIAL SELSYN UNITS.

Many other types of Selsyn units have been developed, including jeweled-bearing units for precision and instrument work, high-accuracy Selsyn units with restricted rotor rotation and special collector leads, and some of extremely compact design to meet limited overall dimensions.

APPLICATIONS.

The Selsyn system in the commercial field has three primary functions: remote signaling, remote control, and transmission of synchronous power. The remote signaling may be audible or visual or both; the remote control, either manual or automatic. Scores of various and ingenious adaptations have been and are being made of both principles. When used primarily to transmit synchronous power, the Selsyn system is rotated continuously at varying speeds, preferably not exceeding 1800 R.P.M., the amount of power transmission being a function of the size of units used and the speed at which the system is rotated. The theory of operation of such a system is complex because of the presence of rotating as well as pulsating voltages, and of acceleration and inertia forces. Such a system requires detailed study before being adapted for a definite duty.—"General Electric Review," September, 1930.

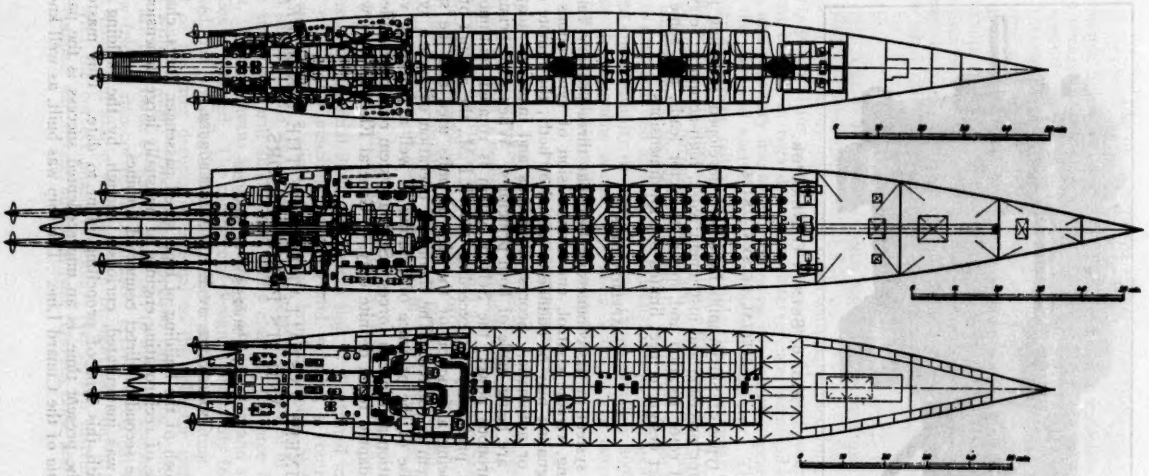
THE MACHINERY INSTALLATIONS OF THE LINER *BREMEN* AND ITS PREDECESSORS.

By PROFESSOR G. BAUER.

TRANSLATED BY E. C. MAGDEBURGER.

The first epoch of the building of high speed passenger liners, characterized by the use of reciprocating engines of unusually large dimensions, was followed by the second of direct connected turbines.

This epoch was inaugurated, curiously enough, by the building of the fastest ship with this type of propelling plant to date. This masterpiece, which up to the present time, is an unprecedented success is the passenger liner *Mauretania* of the Cunard Line. The ship was built, as well known, in



PLAN VIEWS OF MACHINERY INSTALLATIONS.

Fig. 1.—*Mauretania.* Fig. 2.—*Berengaria.* (Ex-*Imperator.*) Fig. 3.—*Ile de France.*

1907 by Swan, Hunter & Wigham Richardson, Ltd., in Newcastle-on-Tyne while its propelling machinery was furnished by Wallsend Slipway and Engineering Co., Ltd., a neighbor of the shipbuilder. The original installation of the machinery and boiler plant is reproduced in Figure 1 to assist the reader's memory.

Saturated steam of 200 pounds (14 at.) pressure is produced by 23 double-ended and 2 single-ended Scotch boilers of usual type with 4 furnaces in each end; these were originally arranged for coal-firing. The total heating surface of these 25 boilers amounts to 14,780 m^2 (159,000 sq. ft.), the grate area of these boilers was equal to 378 m^2 (4,070 sq. ft.).

These boilers are subdivided into four firerooms in such a manner that, with the exception of the forward fireroom, where the ship's form is already considerably narrower, each fireroom has 6 boilers in two rows, with the flue gases of all of these carried to one common uptake. On both sides of the firerooms, along the skin of the ship, were arranged the coal bunkers. The three engine rooms follow the last boiler compartment—two wing engine rooms containing the high pressure turbines driving the wing screws and a larger central engine room where the low pressure turbines of the two central shafts are installed; these shafts also carry the backing turbines.

The arrangement of coal bunkers on the sides of the firerooms and the comprehensive subdivision of the engine rooms by longitudinal and athwartship bulkheads into watertight compartments were chosen with a view to reducing the danger from collision and using the vessel as an auxiliary cruiser; the inconveniences, occasioned by leading steam pipes through the bulkheads, had to be endured as a consequence.

As already mentioned the auxiliary machinery is also arranged in separate compartments adjoining the main engine rooms. A special compartment aft of the main engine rooms contains the condensers; two other compartments aft of condenser room house the circulating water pumps.

The offset of the turbines (high pressure against low pressure) on each side of the ship and the described arrangement of condensers produce a minimum of steam piping and result in minimum of transfer losses; besides the arrangement gave room and suitable location for the installation of the auxiliaries.

The most successful trip with the above described machinery and boiler installation was the 40th outward trip in September, 1910, which took 4 days 10 hours and 41 minutes, which gave for the distance of 2780 sea miles between Queenstown and Sandy Hook a speed of 26.06 knots.

Following the trend of the times the coal firing of the boilers was replaced during the year 1922 with oil firing by the original builders of the ship, which resulted in a considerably improved speed performance of the ship on the Cherbourg-New York route now used. The best trips made after the installation of oil firing were as follows:

		Sea miles	*Days*	*Hours*	*Min.*	*Knots*
—1922	Cherbourg-New York.....	3160	5	9	50	24.35
—1924	Cherbourg-New York.....	3160	5	3	20	25.60
Aug.—1924	New York-Cherbourg.....	3198	5	1	50	26.25
—1928	New York-Cherbourg.....	3160	5	2	34	25.75

But the possibilities of this unusual ship were not exhausted with the above performances. A very rational improvement in the feed water and condenser installations, of which nothing has become officially known, brought further gains in speed, so that the veteran ship only recently, after the first of her seriously to be considered competitors began its career, was capable of the following excellent performance:

	Sea miles	*Days*	*Hours*	*Min.*	*Knots*
Aug., 1929, Cherbourg-New York....	3160	4	21	44	26.85
Aug., 1929, New York-Plymouth....	3098	4	17	49	27.22

Although the *Mauretania* was thus capable of outperforming such imposing competitors in the Atlantic Ocean as the *Berengaria* (ex-*Imperator*), *Leviathan* (ex-*Vaterland*) and *Majestic* (ex-*Bismarck*), it may be added here in all fairness that the last three named ships were built primarily for larger comfort of their passengers than to attempt highest speeds. However, it must not be overlooked that the construction of the *Mauretania* was begun six years prior to the last named three ships and the progress made by engineering science during this considerable period of time was not sufficient to easily overshadow the product of an earlier epoch.

Should we ask ourselves what contributed to the outstanding success of the *Mauretania*, the answer must be, although it is beyond the limits of a paper devoted to propelling machinery, that it was primarily due to an exceptionally happy choice of dimensions and form of the ship's hull.

When the form of the *Mauretania* is now examined critically in the light of the newer researches on the propulsion of ships, we find a number of factors, which—either consciously applied or merely the result of the right technical "feel" contributed to the signal success of the vessel.

Important among these points is the fact that the length of the ship offers the least resistance for its speed from displacement waves.* It may be repeated here that the model towing experiments have shown that this portion of the ship's resistance does not vary continuously with the ship's speed, but contains an additional periodic curve, influenced by the length of the wave, of such type that a relative maximum of resistance obtains when a wave trough is at the stern of the ship, and a smaller resistance must be overcome when at the stern or near it a wave peak forms. Since the velocity of propagation of the displacement waves must be equal to the speed of the ship and the length of the waves is expressed by the formula

$$L = .17V^2$$

where V is expressed in knots, the formation of wave trains around the ship must therefore be in a certain relation to its resistance.

The above shows that it is advantageous to have the length of the ship equal to a multiple of the wave length corresponding to the ship's speed, or even better to be a little in excess of such value.

This relation is clearly shown on the diagram—Figure 4—where the abcissae are the ships' lengths (constant) while the ordinates represent the ships and therefore wave speed (variable), the inclined lines showing the number of wave lengths per ship corresponding to given ship speeds.

It will be seen that these circumstances favor the *Mauretania* as well as the *Imperator* and the *Bremen*, all three ships are a little longer than a multiple of the wave length, so that in addition to the wave peak at the bow there is another somewhere above the propellers. This improves of course also the action of the propellers and reduces the danger of cavitation or sucking in of air into the propeller stream.

The propellers of the *Mauretania* were located comparatively close to the outer skin of the ship and relatively far from the strut besides the inner screws were set about five diameters farther aft away from the wing screws, so that the former work in a stream already quite free from eddies.

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* Compare paper by Dr. Kempf at the 6th annual meeting of the Society of Friends and Patrons of the Hamburg Experimental Basin.

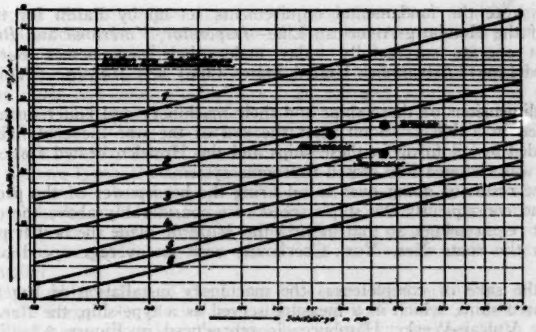

FIG. 4.

Relation Between Ship's Speed (Ordinates) in m/h and Ship's Length (Abscissae) in Meters.

Wellen pro Schiffslänge = waves per ship's length.

The fairwater between the strut and the hub of the propeller consisted of a conical fitting, which according to newer investigations results in increased propulsive efficiency, because the water thereby is guided away from the hub and into more efficient zones of the propellers.*

After exhaustive towing experiments in the model tank of the English Admiralty at Haslar, which developed the most efficient lines of the ship's hull, the builders of the *Mauretania* had a wooden model made to the scale of about 1:16 in order to study the efficiency of the propeller drive.

These very carefully planned experiments showed also the importance of the influence of the wind resistance on the power required to propel the ship, this circumstance has only very recently been again investigated quantatively. The newer investigations (by Dr. Kempf) of this influence resulted in the following formula:

$$W_L = .06v^2F$$

where v is the ship's speed in m/sec. and F the cross section of the ship in square meters. (The liner *Bremen* requires at operating speed and in still air about 1500 horsepower to overcome wind resistance).

The outlines of the upper decks were added to the formerly open model of the *Mauretania* as the results of the above named experiment, and these were designed for least resistance; the bridge, boat and promenade decks became as a consequence well rounded at their forward ends.

Nevertheless the continued up to now success of the vessel would be impossible without the carefully planned and eminently well made machinery and boiler plant, which responded to the increasing demands of the time without renewal of any of the principal parts of the installation.

As already indicated the speed records of the *Mauretania* remained quite safe up to the appearance of the *Bremen* on the Atlantic Ocean. It is true that the tendency in the years immediately preceding the outbreak of the war was toward building very large and comfortable ships for the transatlantic service, of considerable but not necessarily record speed.

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* See Dr. Kempf's paper.

Such were the fundamental requirements set up by Ballin for the new ships of the Hamburg-American Line—*Imperator, Vaterland* and *Bismarck* —which operate successfully today under their new names *Berengaria* (Cunard Line), *Leviathan* (United States Line) and *Majestic* (White Star Line).

The dimensions of these ships and their machinery and boiler installations are given in Table I. Soon after the end of the war the coal firing was superseded by oil firing in these ships and the Howden heated forced draft system was changed into closed firerooms system.

As the result of going over to oil firing the horsepower of the propelling plant and consequently the ship's speed rose considerably above the original contract requirements, so that in 1923 the *Majestic* made the best trip of the year on the route New York-Cherbourg with an average speed of 24.76 knots.

For the sake of completeness the machinery installation of the first of these three ships, which in a measure served as a type-ship, the *Berengaria* built by Vulcan-Werke Hamburg, is reproduced in Figure 2.* Table I gives the principal data of machinery and boilers.

As compared with the *Mauretania* this installation shows a step forward in the replacement of Scotch boilers by water-tube boilers for the purpose of weight saving.

The original operation of these boilers with coal firing was—as observed on all water tube boilers on merchant ships—very difficult and would have caused much trouble in course of time. The large grate areas were difficult to service, the breaking up of clinkers was very slow and cumbersome, pockets of fly-ash would build up among the tubes, which interfered with the passage of the flue gases and created dangers for the boiler tubes themselves; besides it was very difficult to keep an even fire over the whole area of the grates and therefore in some parts of the grate area air would blow by and affect seriously the efficiency of the boilers. Therefore as long as coal firing had to be considered the use of water tube boilers was a pretty dangerous step and the advantages of water tube boilers came to the fore only with the change-over to oil firing.

The difficulties of the water tube boiler were over and another advantage of higher forcing capacity as compared with the Scotch type of boiler became apparent.

Another step forward in the *Berengaria* as compared with the *Mauretania* was to be the successive connection of 4 shafts for ahead operation as compared with 2 on the *Mauretania*. This was done by passing of the steam at full speed ahead first through the H. P. turbine on port center shaft, then through the I. P. turbine on the starboard center shaft and then through the two L. P. turbines on the wing shafts connected in parallel.

Although this arrangement resulted in a better economy at full speed, yet it was considerably more complicated, since for each maneuver such a succession of turbine connections had to be broken up and a new combination established which would change the direction of rotation for either port or starboard pair of shafts.

Along similar lines of thought was built in 1925-26 the passenger liner *Ile de France* for the Compagnie Générale Transatlantique by the French shipbuilders Chantiers et Ateliers de St. Nazaire (Penhoët). Table I gives

* As in the case of the *Mauretania* preliminary experiments prior to the building of the *Imperator* were made with a self-propelled model, which was made geometrically perfect in all hull parts and mechanically as similar as possible, for which purpose the skin, bulkheads and girders were made of brass, and electric motors fed by batteries were used to propel the model.

the principal dimensions and data for the propelling plant of this ship while for completeness sake the plan view of the complete machinery and boiler installation is reproduced in Figure 3.

The turbine plant differs but very little from that of the *Berengaria* class, only the inner and outer shafts have been changed around in the path of steam so that H.P. and I.P. turbines are arranged on the wing shafts while the L.P. turbines drive the center shafts. A deviation occurs in the boilers, since neither Scotch nor water-tube type of boilers, but a combination of the two types, the so-called Prudhon-Capus boiler, was used.

It will be remembered that in this type of boiler the cylindrical part of the Scotch boiler is only as long as the furnaces while the combustion chambers are formed by nests of water tubes which are placed between the upper and lower drums, which in turn are riveted to the rear wall of the cylindrical boiler.

Mostly double boilers of the Capus-type were used in the *Ile de France*. They were formed by placing two single-end boilers end to end and separating the rear combustion chambers by a wall of fire brick, so that the two halves were independent and firing of one end operated only the corresponding half of the boiler.

While even in 1924 the decision over the type of the propelling machinery for at that time the largest ship under construction—*Ile de France*—was made in favor of direct driving turbines, for smaller propelling plants and for naval propulsion the high speed turbines with gear drive were already successfully used for a long time.

On December 13, 1926, were ordered the two passenger liners *Bremen* and *Europa* and it may be well stated that with this step, showing utmost foresight and courage on the part of the North German Lloyd, a new epoch in the building of high speed passenger liners was begun.

The status of engineering of that day determined the principal characteristics of the propelling plant as follows: subdivision of the total horsepower into four separate machinery units, high speed turbines with gear drive and water tube boilers with oil firing, using steam of a considerably higher pressure and superheat as compared with previous practice.

Before going into details of the machinery installation some remarks of general interest concerning the building of the propelling plant of this ship may not be out of place.

The building period of the ship specified in the contract—27 months—was extraordinarily short; nevertheless it was possible to meet it by careful planning and the ship's delivery was delayed only by the duration of a three months long strike of all branches of the shipbuilder's works.

The more important dates were as follows:

Date of order	Dec. 13, 1926.
Installation of boilers, which to expedite matters was done prior to launching	Feb. 9-April 17, 1928.
Tests of turbines in the shops	July 19, 1928-Feb. 11, 1929.
Launching	August 16, 1928.
Complete suspension of work due to strike	Oct. 1, 1928-Jan. 5, 1929.
Firing up of the first boiler	April 17, 1929.
Dock trials of turbines	May 17-June 11, 1929.
Transfer of the ship from Bremen to Bremerhaven	June 24, 1929.
Preliminary trials	June 27-28, 1929.
Contract trials	June 29-July 6, 1929.
Beginning of maiden voyage to New York	July 16, 1929.

The maiden voyage of the ship was crowned by complete success and brought the purchaser and builder the assurance that both ship and machinery were satisfactorily designed. On the first outward trip and the following return trip the *Bremen* broke all existing speed records of passenger liners in the Atlantic Ocean and thus won the coveted "blue riband of the ocean." It is true that the liner *Europa* on its maiden voyage to New York about nine months later improved the speed record by about .08 knot, establishing on that trip a speed of 27.91 knots, or exactly the same speed (up to two decimals) as was made by the *Bremen* on its first return trip eastward.

Thus it may be said that neither possesses the "blue riband" unchallenged, or both of them have it jointly, namely one for the westward, the other for the eastward crossing.

The speed results of the *Bremen* for the first year of operation are compiled in Table 2. This table shows better than any statement the reliability with which the propelling plant has performed during that period.

The problem presented to the designer of the propelling plant for this passenger liner was the more responsible because the North German Lloyd had almost no experience with marine turbine installations. At the time the *Bremen* was ordered the fleet of this company included only the following turbine driven ships: S. S. *Koblenz* built in 1924 of 2 × 1950 S.H.P., the single-screw freighter *Porta* (1923) of 1600 S.H.P. and the seashore passenger ship *Roland* commissioned in 1927 with a 2 × 2150 S.H.P. These were only very small ships of small power and could hardly be of any assistance in the design of the *Bremen*.

It is therefore the more to be wondered at and shows the brilliancy of the organization of the North German Lloyd, that they were nevertheless in a position to quickly take over the operation of the new vessel, which was done right from the start in the most excellent and praiseworthy manner—an accomplishment on which the technical director of the North German Lloyd—Herr Director Koch—may surely be congratulated.

The unusually rapid completion of the propelling plant of the *Bremen* would have been impossible if it were not for the merger of the two great shipbuilding organizations, A. G. Weser and Vulcan of Hamburg, which occurred shortly before the order was obtained and brought together under my direction a number of engineers particularly experienced in the building of machinery for high speed steamers. Among these may be named first of all C. Schmieske and H. Böhm, of which the first was responsible for the design of the complete main propelling plant including turbines, gear drives, boilers, etc., while the latter was entrusted with the required theoretical computations, especially for turbines, also shop tests and measurements on board ship later. Messrs. J. Wolfenstetter and R. Kabelac bore the responsibility for providing and designing the auxiliary machinery, including auxiliary Diesel generator plant, as well as such arrangements, which are along the border line between hull and engineering departments and which in ships of this type are of considerable importance, such as steam heating, refrigeration, fire service, etc.

L. Schnitger was responsible for the manufacture of the complete installation in the shops, while A. Holst, with his long experience in the assembly and starting of the largest machinery, could utilize his experience most effectively. Mr. Holst remained on the ship throughout the first year of its operation and assisted the personnel of the North German Lloyd to acquire their experience, which circumstance no doubt contributed materially to the successful operation of the ship throughout the year of guarantee period.

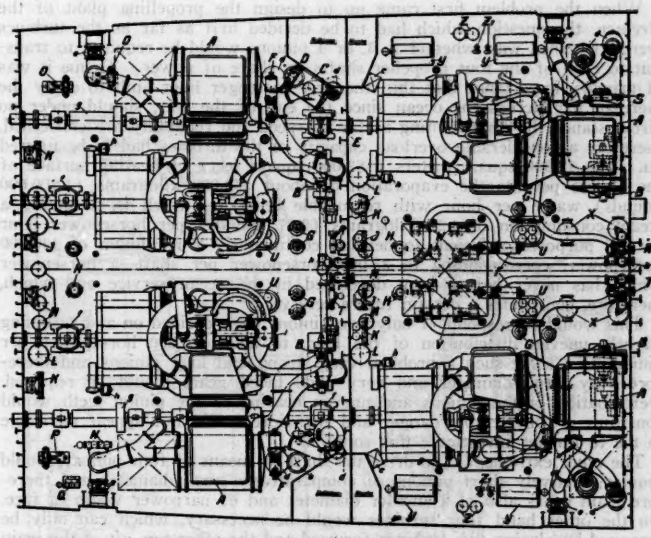


FIG. 5.—PLAN VIEW OF MAIN ENGINE ROOMS,

LEGEND FOR FIGURES 5-6-7

A-Main condensers.
B-Main Circ.water pumps.
C-Auxiliary condenser.
D-Wet air pump for aux. condenser.
E-Circ.water pump for aux. condenser.
F-Steam jet air pump, 1st and 2nd stage.
G-Condensate and drainage pumps.
H-Main lub. oil pumps.
J-Reserve lub.oil pumps.
K-Circ. water pumps for main lub. oil coolers.
L-Main lub. oil coolers.
M-Lub. oil filters.
N-Deck wash pump.
O-Bath pumps.
P-Sanitary pumps.
Q-Fire Pumps.
R-Fresh water pumps.
S-Ballast pumps.
T-Bilge pumps.
U-Drain tanks of turbines.
V-Hot well.
W-Main feed pumps.
X-Feed water heaters.
Y-Evaporators.
Z-Feed water heaters for evaporators.
Z_1-Brine pump for evaps.
a-Bilge water oil recovery.
b-Cloth filter for bilge water oil recovery.
c-Lub.oil preheating tank.
d-Sea valves for main cir. water pumps.
e-Blowout gate valves for main cir. water pumps.
f-Thrust bearings.
g-Torsion meter.
h-Turning gear for the main turbines.
i-Maneuvering stand.
k-Bulkhead valve group of steam piping.
l-Emergency bilge valve.

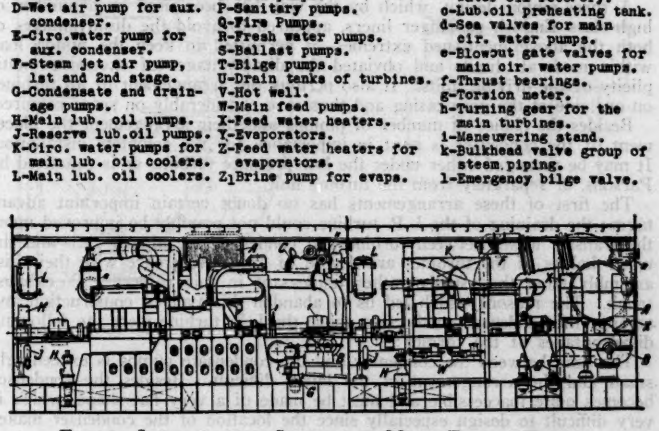


FIG. 6.—LONGITUDINAL SECTION OF MAIN ENGINE ROOMS.

When the problem first came up to design the propelling plant of the *Bremen*, the question which had to be decided first as far as the turbines were concerned was whether 2, 3, or [illegible] mit to each [illegible] of [illegible] German [illegible] circumstances be disappointing as far as [illegible] and reliability are concerned, therefore a considerable overload capacity of the turbines had to be figured on. The 17,000 square meters (183,000 square feet) of heating surface of the boilers permits the evaporation of about 550,000 kilograms (1,200,000 pounds) water per hour with reasonable economy, which is, assuming a steam consumption of 4.25 kilograms (9.37 pounds) per horsepower hour for all purposes, sufficient to develop continuously a maximum of 140,000 horsepower, corresponding to 35,000 horsepower per shaft—a horsepower which has never before been developed in the ocean service and which, therefore, constitutes a certain future reserve.

This would correspond, if only two pinions were decided on and assuming slightly uneven distribution of the load, to about 20,000 horsepower per pinion. To solve such a problem very strong and long pinions and therefore very large diameter and very wide faced gears would be required. Deformation of the pinions and uneven bearing of the pinion teeth would constitute a considerable danger and therefore it did not appear advisable to me to further investigate this solution.

The other extreme, *i.e.*, to drive the shaft by means of four pinions, would require relatively short pinions of comparatively small diameter and therefore main gears also of a smaller diameter and of narrower width of face. On the other hand four turbines would be necessary, which can only be arranged by placing two turbines forward and the other two aft of the main gear wheel. Such an arrangement makes the units relatively long, with a large number of bearings, glands and transfer pipes, some of which must be led over the gear casing. The design of the gear casing when using four pinions possesses certain disadvantages as far as dismantling accessibility is concerned.

A three-pinion design, which by the way has been used on a number of high-class English passenger liners, appeared to avoid the disadvantages of both the above mentioned extremes—it permitted to keep the pinion load within moderate limits and obviated the disadvantages of excessive multiplicity of the driving units. It also permits the arrangement of all turbines on one side of the gear casing and thus save considerably on space required.

Besides the choice of number of pinions and their arrangement the placement of the condenser is next in importance for the complete installation. It may be arranged either under the low pressure turbine, as is practiced by Parsons, or separately from the turbine unit.

The first of these arrangements has no doubt certain important advantages: the draining of the L.P. turbine could not possibly be improved upon, the transfer losses between turbine and condenser are almost zero and the upper halves of the turbines are devoid of any interference with their disassembly from the usually rather unpleasant in this respect L.P. exhaust trunk. The reasons which led us to abandon this type of construction and arrange the condenser on the side above the L.P. turbine were the following disadvantages of the former:

The head between the condenser and the condensate pumps is exceedingly small and the arrangement of the latter difficult. Besides the condenser becomes quite inaccessible and must be made of a very flat shape, which is very difficult to design especially since the location of the condenser makes it more difficult to provide a sufficiently rigid foundation for the low pres-

FIG. 7.—FORWARD MAIN ENGINE ROOM BULKHEAD.

FIG. 8.—H. P. TURBINE.

sure turbine. These difficulties increase quite considerably for the sizes of condenser and L.P. turbine, such as were required for the *Bremen*. As a further disadvantage of the placing the condenser below the turbine is the fact that beginning with a certain output the vertical rake of the shaft becomes unwieldy and the center of gravity of the turbine unit is pushed too far upward, thus affecting unfavorably the stability of the vessel.

After giving the principal reasons, which determined the arrangement of turbines and gear drive, I would like to consider in detail these units (see Figures 5-7).

Each propeller shaft is driven by a large gear wheel of 4087 millimeters in diameter, which in turn is driven by three pinions of 414 millimeters in diameter; a turbine speed of 1800 R.P.M. corresponds therefore to a propeller speed of 182.5 R.P.M. To provide the best possible foundation, the grouping of the turbines was so chosen that the H.P. turbine was placed near the top of the gear wheel so that the upper parts of the I.P. and L.P. turbines could be dismantled upward, the axes of the I.P. and L.P. turbines were placed in the same horizontal plane with the centerline of the propeller shaft.

In the forward engine room, *i.e.*, on the two wing shafts, the L.P. turbine is placed inboard, while on the two center shafts it is placed outboard, such an arrangement giving the best utilization of the space available as can be easily seen from the illustrations.

Figure 8 shows a partial section of the high pressure turbine. The rotor consists of a Curtis impulse type wheel of 1200 millimeters mean blade circle diameter, which is followed by a drum with 30 stages of reaction type blading. Steam is distributed by four nozzle manifolds, of which two are above and the other two are below the horizontal axis, and these manifolds are not set-in direct into the turbine housing but the nozzles are fastened in separate hollow housings comprising about one-quarter of the circumference each. The impulse stage disc is shrunk onto the rotor, which is made of a one-piece forging. The housing is of cast steel. Particular care was taken to provide an especially rigid flange connection between the upper and the lower halves of the housing; in operation so far even under overload these horizontal joints remained perfectly tight.

The steam supply through the nozzle boxes of the impulse stage is sufficient to develop a total of about 115,000 S.H.P. When it is desired to use more steam, *i.e.*, to increase the horsepower developed still further, a by-pass valve must be opened which will admit fresh steam to the rear of the impulse stage direct to the reaction blading of the drum.

The glands consist of a number of rings with sharp edges which are carried in special split housings and are made of a composition especially suitable for the purpose. Carbon ring stuffing boxes precede the above sharp-edged rings and a suction chamber for steam leakage is arranged between these two.

The H.P. turbine housing rests with its forward end on a very substantial cast iron pedestal which straddles the forward bearing of the gear wheel and is bolted on to the foundations between the other two turbines, thus making the bearing accessible. The aft end of H.P. turbine housing rests in a very wide bracket substantially secured to the gear wheel housing.

The housings of the H.P. and I.P. turbines are held securely by bolts through their aft feet to the foundations while the forward foot can glide freely. For this purpose the forward foundation bolts are not tightened and a plate of special brass composition is placed between the foundation and the foot to permit easier gliding.

The L.P. turbine housings are held to the foundation in their middle leaving the ends free to glide.

The following may be said with regard to heat expansion of the rotors: Figures 8 to 10 show that each turbine rotor has a single collar thrust bearing of a well-known type at its forward end, which serves to take the difference in thrust of the bladed drums and the dummy piston and in the L.P. turbines the inequality of the thrust of the two halves of the turbine. This thrust collar at the same time serves as the point of support of the rotor with relation to the housing. The rotor thus glides in each case aft with respect to the housing and the axial clearances of the bladed rows were so dimensioned after careful investigations of all the possibilities of relative displacement as to take care of the influence of temperature difference between rotor and housing.

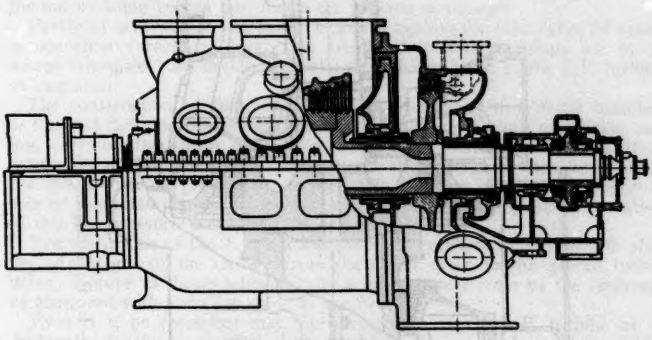


FIG. 9.—I. P. TURBINE.

Returning now to the consideration of the main turbine unit. The intermediate pressure turbine is shown on Figure 9. The section shows the last rows of blades which constitute the second group of forward blading. On the right side of the illustration is shown the section of the high pressure backing turbine consisting of a disc with three rows of impulse blading having a mean blading diameter of 1400 millimeters, which disc is shrunk on direct onto the shaft of the rotor, which in turn is connected to the drum itself by a heavy flange. Ahead and astern turbines are separated by a hollow head and a labyrinth packing.

The exhaust of the ahead turbine is carried off by a flange connection on top of the housing while the astern turbine exhausts into the L.P. turbine through a flange on the bottom.

The L.P. turbine—shown in Figure 10—is bladed for ahead and astern operation in accordance with the double-flow principle.

The rotor of the L.P. turbine consists of a drum of 1000 millimeters in diameter which carries the astern blading; to both ends of this drum are fastened by substantial flanges end pieces which carry the journals and the blading of the ahead turbine. To provide additional safety against centrifugal stresses the whole rotor is made of chrome-nickel steel of 60 to 70 kg/mm^2 (85,000-100,000 lbs/in^2) tensile strength, 18 per cent elongation and about 40 to 50 kg/mm^2 (57,000-71,000 lbs/in^2) elastic limit. Since maximum stress in normal operation does not exceed 1500 kg/cm^2 a factor of safety of four is thus provided.

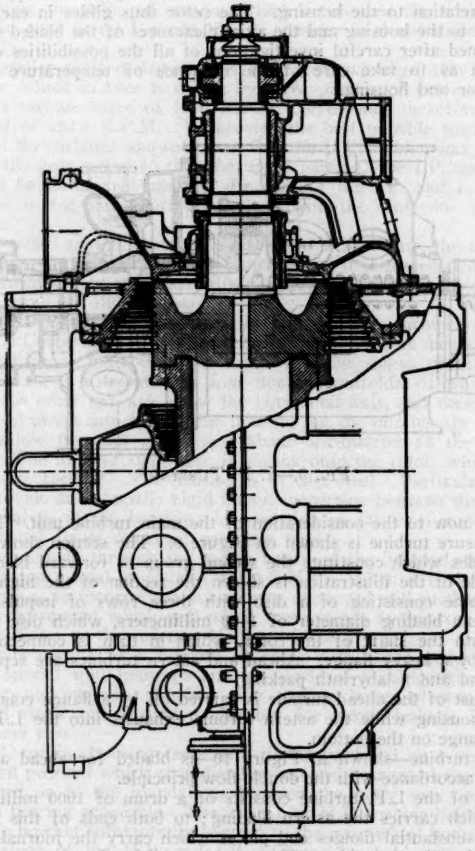


FIG. 10.—L. P. TURBINE.

The leading thought which governed the design of this drum was to use only such forgings which permit assurance of their soundness; this was accomplished by using end pieces of very simple shape, which could be forged both axially and radially, and a comparatively thin walled drum.

A detail which seems worth mentioning is the method of joining the drum to the end pieces, which is shown on Figure 11. The studs screwed into the end pieces have a collar to prevent the backing up of these studs and the nuts are of the closed type, whose cylindrical body projects deeply into the flange of the drum and for the purpose of taking the bending stresses caused by the centrifugal force are surrounded by a ring, fitted into the bore in the flange. The locking of these nuts against turning is done by means of this fitted ring which is held in the bore of the flange by means of two pins and holds the nut by being peened into six of the grooves in the nut.

Partly to save weight and partly to guard against the occurrence of cracks in operation (cracks in large cast iron L.P. turbine housings are by no means infrequent) we decided to make the housings even of the L.P. turbines of cast steel.

The construction of these housings shows an interesting detail inasmuch as the part carrying the blading of the astern turbine is made a separate casting, carrying the steam connection at the very bottom, which can freely expand in all directions. This safety precaution appeared necessary because the steam entering L.P. astern turbine still carries a considerable superheat (up to about 210 degrees C. steam temperature) and therefore deformations of this low pressure part had to be guarded against.

The discussion of the L.P. turbine gives an excellent opportunity to show the advantages of the change from the direct drive to the geared turbine drive. Figure 12 shows schematically a low pressure rotor of the *Imperator* as compared with a similar rotor of the *Bremen*.

Even if it be remarked that the horsepower of the L.P. turbine of the *Imperator* develops a quarter of the total horsepower or about 15,000 S.H.P. while on the *Bremen* the L.P. turbines develop only one-twelfth of the total or only about 7500 S.H.P., nevertheless the rotor of the *Bremen* carries in addition to the two sets of blading of the ahead turbine arranged for double-flow on the ends also the blading of the astern turbine on the center drum. If in addition we remember that the rotor of the *Imperator* carries a total of 55,530 blades while *Bremen's* rotor has only 10,888 blades, and the steam consumption of the *Imperator's* turbines is 5.3 kilograms per S.H.P./hour (11.66 lbs. per S.H.P./hour) while those of the *Bremen* require only 3.5 kilograms per S.H.P./hour (7.72 lbs. per S.H.P./hour), the mighty progress made by engineering becomes evident.

In order to illustrate the path of steam in the turbine installation we will use the so-called J-S or Mollier diagram (Fig. 13), the ordinates of which show total heat while entropy values are plotted as abscissae.

This diagram is based upon the assumption of 3 atmospheres (42.6 lbs. per sq. in.) drop in pressure between the boiler and the turbine, *i.e.* from 24 atmospheres to 21 atmospheres, as well as upon a steam entering the turbine at a temperature of 350 degrees C. The back pressure in the condenser is assumed to be equal to an average value of .05 atmospheres (28.5 in. vacuum).

The adiabatic expansion in the impulse stages of the H.P. turbine is represented by the vertical line AB; the efficiency of the wheel is 69 per cent so that the heat drop BB' is lost. The losses are transformed into heat so that the final condition of steam passing through the impulse stages is represented by point C. In a similar manner the adiabatic drop for the H.P. drum is shown by a vertical line CD of 59 WE or Kcal (234 B.T.U.'s) but due to

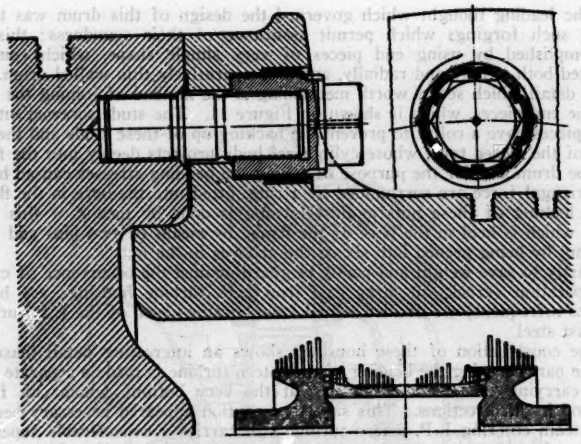


FIG. 11.—TYPE OF LOCK USED IN L. P. ROTOR JOINTS.

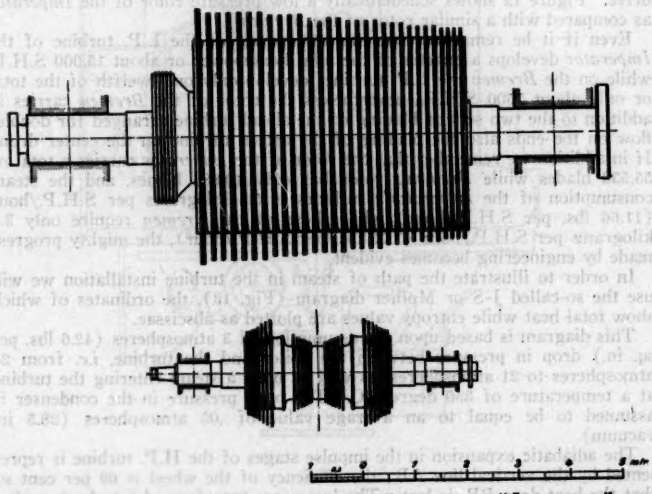


FIG. 12.—COMPARATIVE SIZE OF L. P. ROTORS OF "IMPERATOR" AND "BREMEN."

indicated thermodynamic efficiency of 78.5 per cent the passing of H.P. drum will be represented by the point E.

Taking into account small wire drawing losses in the transfer pipe the initial condition for the intermediate pressure turbine will be represented by point F. The first part of the I.P. drum with an efficiency of 80 per cent will give a final condition represented by point H while the second part having an efficiency of 77 per cent a final condition K. Friction losses in the transfer of steam to the L.P. turbine will make the initial condition for the latter to correspond with point L. The adiabatic drop LM of the L.P. turbine will be transformed into mechanical energy equal to LM' with an efficiency of 70.8 per cent and the final condition of the steam exhausting out of the L.P. turbine will be represented by point N corresponding to a vacuum of 95 per cent (28.5 ins.).

The same diagram has been used to represent the path of steam for astern operation. The adiabatic drop through the impulse stages of the astern turbine, consisting of a Curtis wheel attached to the shaft of the I.P. ahead turbine,—is represented by the vertical line PQ, with an efficiency of 48.8 per cent corresponding to line PQ' the final condition of steam can be represented by point R. Due to friction losses in the transfer pipe the initial condition of steam entering L.P. astern turbine is represented by point S and, using 91 per cent vacuum, the adiabatic drop can be represented by the vertical line ST which is transformed into mechanical energy ST' with an efficiency 55.7 per cent. The point U represents the final condition of steam when backing.

From a weight of steam equal to 90,000 kilograms (198,000 lbs.) per turbine unit per hour the process represented on this diagram gives 103,600 horsepower for the whole ship and a steam consumption of about 3.5 kilograms (7.7 lbs.) per horsepower hour as was already mentioned. The combined efficiency of the ahead turbines including their frictional losses, however, without taking into account the gear drive is according to the above calculation 76.3 per cent, but in actual operation efficiencies up to 79 per cent were measured, although with higher steam temperatures in the turbines.

Despite the introduction of high speed turbines with gear drive and the consequent reduction in the number of blades and turbine dimensions as convincingly shown in Figure 12, turbine casualties occur even now quite frequently although they lost much of the dread of the former years due to smaller number of blades and reduced turbine dimensions, it was therefore one of my principal worries in the design of such an important ship to try to prevent their occurrence. The liner *Bremen* fortunately up to now has been saved the consequences of turbine failures.

The following points demand important consideration to secure successful results:

(*a*) Selection of correct material for the turbine blading;
(*b*) Correct design and substantial dimensions of blading;
(*c*) Correct axial and radial clearances;
(*d*) Absolutely reliable drain arrangements;
(*e*) Reliable lubrication;
(*f*) Safeguards against entrance of foreign objects;
(*g*) Prevention of blade vibration and torsional critical speeds of the vibrating system;
(*h*) Careful design of rapid shut-off devices.

The following may be said to amplify these points:

As *blade material* was used Monel metal throughout with the exception of impulse wheels, the blades of which were made of rust-resisting steel. All guide vanes were also made of rust-resisting steel.

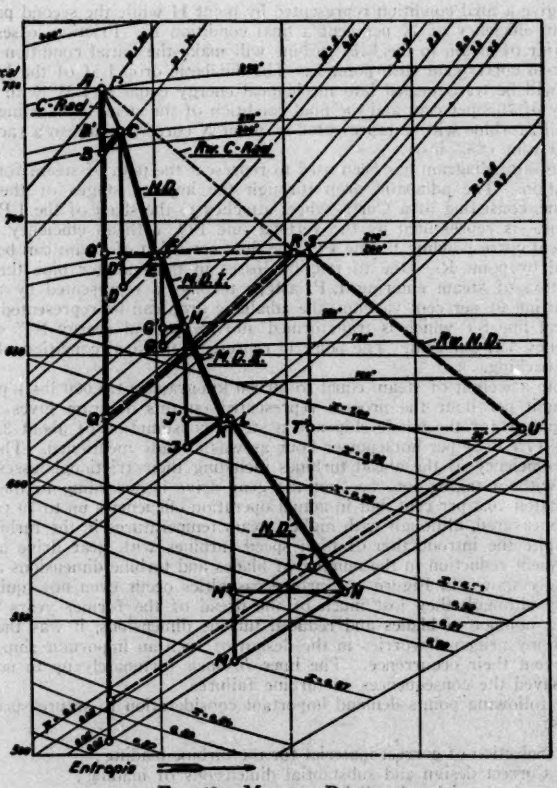


FIG. 13.—MOLLIER DIAGRAM.

Example:

G = 90,000 Kg./hr.
N = 103,600 S.H.P.
p_4 = 21 atmos.
t_4 = 350 degrees C.

To secure absolute certainty, as far as *tightness of blading* is concerned, blades milled out of heavy sections were used extensively, the heaviest stressed blades were made one piece with the filler-piece, while the less stressed blades, although also milled out of heavy sections, had still a narrow filler-piece between them.

With the exception of the impulse wheel blades all blading was bound together in a well-known manner by means of bi-metal wire and sharpened at the tip and only impulse blading was securely fastened to shrouding.

To prevent *friction between rotor and stator blading the* relative motion between them due to different expansion of rotor and housing under heat was carefully calculated. When such a rubbing occurs and does not become immediately noticeable, there is still the great danger that when the direction of rotation of the turbine is altered the spring of the blading will lead to interlocking and result in complete stripping of all blading. As already mentioned the point of anchorage of the relative position of the stationary and rotating blades is in the thrust bearing of each turbine. The thickness of the white metal of this thrust bearing is so dimensioned that even should the white metal melt out there still will be no contact beween the blades.

To secure correct *radial clearance*, the careful warming-up of the turbines when starting them is of course of prime importance, since otherwise the rotors begin to vibrate and may develop contact of blades with the housing in radial direction. In order that such contact, should it nevertheless develop due to some unfavorable circumstances, does not lead to immediate destruction of blading, each individual blade is sharpened at the end in a well-known manner, so that only a small amount of heat may develop and the distortion of individual blades or of the housing becomes exceedingly improbable.

Damage to turbines and especially such as lends to destruction of blading even in modern times is frequently due to *water impacts*, which result from unsatisfactory drainage arrangements or also through *priming* of boilers. In the *Bremen* propelling plant extraordinary precautionary measures have been taken in this respect. First of all there is in the main steam pipe a liberally dimensioned water-collector on the forward bulkhead of the engine room; furthermore, wherever necessary there were cast on substantial drain-pots at the turbine housings, which conduct the condensate of each turbine unit into a common tank, from where it drains into a special drainage pump which is in one housing with the condensate pump.

Most failures due to *water impacts* usually occur, as well known, while operating astern, when the weaker reaction blading of the astern turbines is bent and catches when direction of rotation is altered. In the case of the *Bremen* the conditions are so far favorable as the steam of the high pressure astern turbine enters the intermediate pressure ahead housing at its lowest point and must exhaust into the chamber of the low pressure astern turbine upward. The larger part of this transfer pipe, particularly below the flange of the L.P. turbines, is equipped with a specially designed strainer grating which should serve to intercept and keep in the lower part of the pipe the water which was for one reason or another carried thus far. This serves to materially reduce the danger from water when running astern.

Modern engine building art lays great stress on a *reliably operating lubricating oil system* and it becomes the more important the higher the speed and the output of the plant. In designing the *Bremen* particular stress was laid on lubrication of all bearings. First of all the lubricating oil pumps were liberally dimensioned. In addition to a separate main lubricating oil centrifugal pump capable of delivering 270 m^3 (71,400 gallons) per hour a reserve lubricating oil pump of plunger type of one-third the capacity of the main lubricating oil pump was provided for each unit. In addition to the

normal lubricating oil filters and coolers, which are supplied with each installation, a reserve unit is furnished for each two of the turbine groups. The lubricating oil piping is also liberally dimensioned. The oil supply to each individual outlet may be observed from the operating stand through glass walled fixtures and each bearing is of course equipped with a thermometer. A detailed description of the lubricaitng oil distributing system follows under the description of the reduction gear.

To prevent introduction of foreign objects steam strainers are provided in the main steam pipes immediately before the blading, *i.e.*, before each nozzle box and in the bypass for forced speed, as shown on Figure 8.

The water collecting tank serves also to separate carried away parts, such as parts of valves, etc., out of the main steam piping, which in itself is so designed that all inner ribs or other protruding obstructions are avoided. This tank is equipped with a heavy deck plate above its water compartment so that no hard objects or even water can again be carried upward.

The computation of blade vibration offers as before considerable difficulties both in the theoretical calculation itself and in properly evaluating the practical conditions, so that for instance a check of a failure from vibration with resultant destruction of blading can hardly be made reliably. Systematic experiments on excitation of blade vibration, which were made already many years* ago at the Vulcan-Werke, Hamburg, resulted, however, in a method of calculation, which permits determination of frequency of free vibration of single blades as well as whole segments or groups of them with results quite satisfactory for all design requirements.

These calculations were applied to the turbine blading of the *Bremen* and in order to be doubly sure experiments were made in the shops on bladed rotors to excite free vibration of the bladed parts. These experiments were made as those previously described by directing a stream of compressed air against the blading which stream was interrupted by a rotating disc with a hole, the speed of which could be regulated, and converted into periodic impact impulses. In such a manner it is very easily possible to determine the frequency of free vibration of the blading with great accuracy and thus check the results of calculation.

A special series of experiments were carried out to determine the factors which have a damping effect upon vibration amplitudes. It was shown that free blades could be excited by the smallest impulses even if the frequency of the exciting impulses was only a fraction of frequency of free vibration. With the addition of binding the amplitudes of vibration become smaller and smaller and they finally can only be excited at critical speeds corresponding to synchronism with their frequency of free vibration. This was one of the reasons—there were quite naturally other considerations—why the blading of the *Bremen's* turbines was equipped with heavy binding by means of strong wire.

The *calculation of the frequency of torsional vibration of the main propelling unit* showed that shafting dimensioned in accordance with the requirements of the classification societies would have the one noded frequency of only 170 vibrations per minute or only 10 revolutions below the normal operating speed. By increasing the size of the shafting in the actual installation this natural frequency of torsional vibration was raised 225 V.P.M. and thereby definitely removed beyond the operating range. It may appear excessive to worry about frequencies of torsional vibration of a turbine installation which has such an even turning moment. This objection appears quite justified, generally speaking, but on the other hand one can hardly object to such an easily attainable additional safety in a very costly installation.

* See Author's paper in Jahrbuch der Schiffbautechnischen Gesellschaft 1911.

	Propeller WR^2. kg m^2	Reduced length of shafting cm	Gear Drive WR^2 kg m^2	Red.length of turbine shafting cm	Turbine WR^2 kg m^2	Turbine
A	126,500	6,050	455,000	453.5	1,450	H.P.
				464.8	4,020	I.P.
				408.8	19,700	L.P.
B	126,500	6,050	455,000	4.65	141,000	H.P.
				4.77	392,000	I.P.
				4.20	1,920,000	L.P.

Lengths are referred to a polar moment of inertia = $10^5 cm^4$

Gear Ratio = 9.87

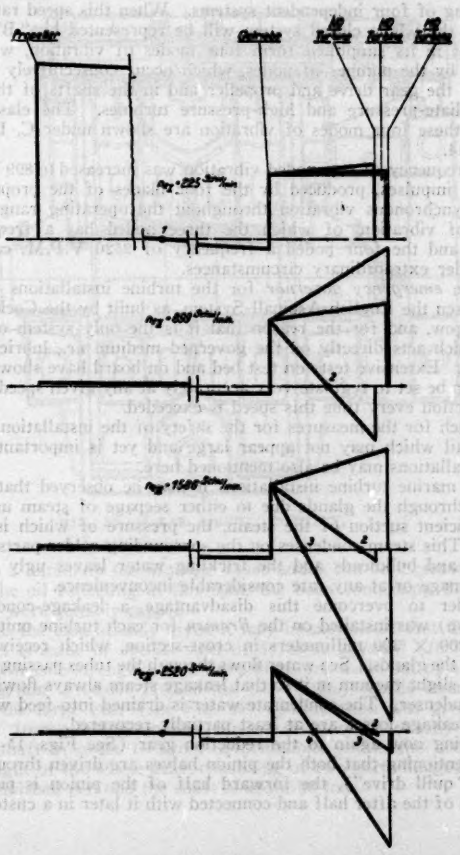


FIG. 14.—TORSIONAL VIBRATION DIAGRAMS FOR CENTER SHAFTS SHOWING LOCATION OF NODES AND RELATIVE AMPLITUDES.
Setriebe = Gear drive: N.D. = L.P.; H.D. = H.P.; M.D. = I.P.

Turbine propelling plants may nevertheless be excited torsionally, although to a lesser extent, primarily by the blades of the propeller, which, due to the variable wake of the propeller circle, superimpose a variable torque over the constant torque of the turbines. Besides inaccuracies in the pitch of the gears and pinions can also excite torsional vibration, since each irregularity of torque may be subdivided into a series of its harmonic components. Uneven steam impact on turbine blading, as well as eccentricities of some parts may act in the same way, quite independently from the fact that casualties, such as damage to one of the propeller blades, may increase the exciting force very heavily.

The torsionally vibrating system, representing the main propelling unit of the *Bremen* is shown in Figure 14 by "A," reduced to one common diameter of shafting and to a common radius of gyration for all rotating masses, however, without taking into consideration the speed ratio of the gear drive, *i.e.* consisting of four independent systems. When this speed ratio is taken into account the whole closed system will be represented by "B". This system possesses in its simplified form four modes of vibration, which are distinguished by the number of nodes, which occur consecutively in one shafting between the gear drive and propeller and in the shafts of the low-pressure, intermediate-pressure and high-pressure turbines. The elastic lines representing these four modes of vibration are shown under C, D, E, and F on Figure 14.

The frequency of two-noded vibration was increased to 899 V.P.M. in order that the impulses, produced by the four blades of the propeller, could not excite synchronous vibration throughout the operating range. The higher modes of vibration, of which the three noded has a frequency of 1586 V.P.M. and the four noded a frequency of 2520 V.P.M. could be excited only under extraordinary circumstances.

As the *emergency governor* for the turbine installations of the *Bremen* was chosen the English Aspinall-System, as built by the Cockburn Company of Glasgow, and for the reason that it is the only system of governing to date, which acts directly on the governed medium *i.e.*, lubricating oil under pressure. Extensive tests on test bed and on board have shown that the governor can be set to regulate very accurately at any given speed in R.P.M. and will function every time this speed is exceeded.

So much for the measures for the safety of the installation.

A detail which may not appear large and yet is important for such turbine installations may be also mentioned here.

In all marine turbine installations it may be observed that a little steam escapes through the glands due to either seepage of steam used for sealing or insufficient suction of the steam, the pressure of which is above atmospheric. This steam condenses on the surrounding colder parts of machinery, gratings and bulkheads and the trickling water leaves ugly rusty paths or other damage or at any rate considerable inconvenience.

In order to overcome this disadvantage a leakage-condenser (patent Schmieske) was installed on the *Bremen* for each turbine unit, a rectangular box of 200 × 300 millimeters in cross-section, which receives the leakage piping of the glands. Sea water flows through the tubes passing in this box and creates a slight vacuum in it so that leakage steam always flows automatically to the condenser. The condensate water is drained into feed water tanks and thereby leakage losses are at least partially recovered.

Returning now again to the reduction gear (See Figs. 15-17) it may be worth mentioning that both the pinion halves are driven through the central flange ("quill drive"), the forward half of the pinion is pushed over the extension of the after half and connected with it later in a customary manner.

I consider this design advantageous because the toothed portion of each of the pinions is subjected only to a maximum at one-half of the total torque being transmitted. The total twist of the pinion over the total width of the teeth is reduced to an exceedingly small portion of it, which naturally has a very beneficial effect on gear noise and length of life of the teeth.

The central shaft extending out of the forward end of the pinion is tapered and carries the spider of a jaw coupling—Figure 18. A similar spider is attached to the end of the turbine shaft and over both spiders the two sleeves of the coupling are glided into place and bolted by their central flanges.

Jaw couplings of this type permit, as already well known, a small relative motion, axial as well as radial, of the turbine and pinion shafts. Since all three pinions are in all respects alike the three couplings could also be made duplicates of one another.

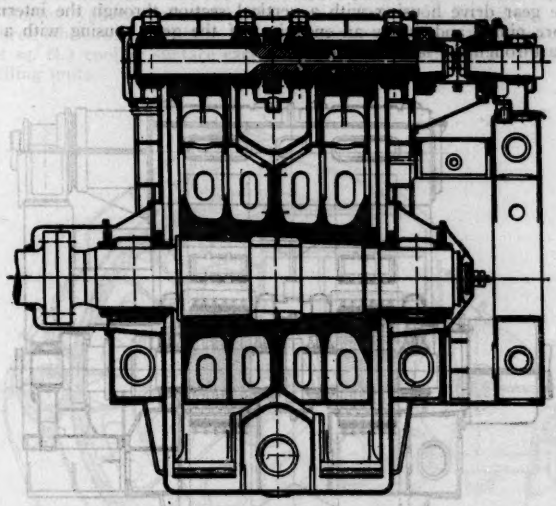


FIG. 15.—GEAR DRIVE. SECTION.

To reduce to a minimum the windage losses which can be increased by the resistance of the oil collecting at this spot, the bolts connecting the flanges of the coupling were suitably capped.

In addition to the data already given it may be mentioned that the rim of the large wheel is made of special steel of 55 to 62 kg/mm² (78,000-88,000 lbs. per sq. in.) tensile strength while the pinions are made out of alloy steel of 70 to 77 kg/mm² (100,000-110,000 lbs. per sq. in.) tensile strength. In the design of the large gear (see Fig. 15) consideration was given to the distribution of forces acting on the rim to disks made exactly at right angles to the shaft in order to avoid deformations within the body of the gear. The weight of the whole gear, consisting of two parts flanged together and including finished rim and the shaft is about 60 tons.

The gears were cut in the shops of the A. G. Weser on a large new gear-cutting machine, which made it possible to produce the shape of the teeth with extraordinary accuracy. Immediately after the last cut the teeth were examined by means of the most accurate optical instruments with regard to pitch variations, which resulted in the proof that the cutting of the teeth was done with an accuracy heretofore unattainable, since the deviations measured amounted to only one quarter of those measured by the same instruments on gears of smaller size which were cut on other also very good gear-cutting machines.

In a similar manner, as described above, were also examined for inaccuracies of pitch the pinions, which were cut on a special gear cutter in the shops of the Vulcanwerft-Hamburg.

Figures 15-17 show a longitudinal section through the main gear and the H.P. pinion, assumed slightly displaced into the vertical plane, an elevation of the gear drive housing with a vertical section through the intermediate pressure pinion and finally an end view of the gear housing with a cross-section through the large gear.

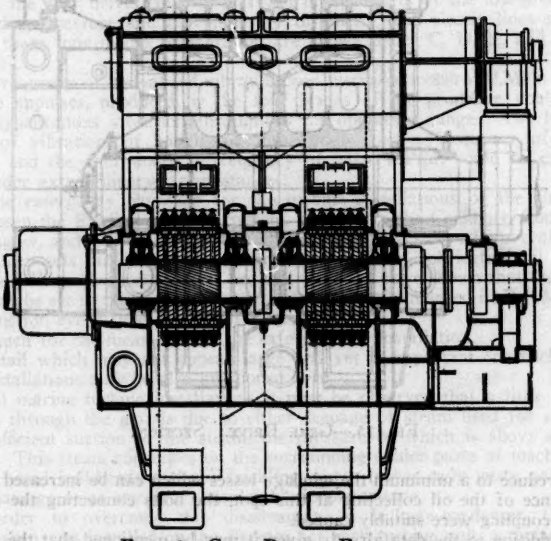


FIG. 16.—GEAR DRIVE. ELEVATION.

Each pinion has four bearings to support it as shown on Figure 15, the two outer bearings in the forward and aft walls of the gear housing and the center bearings in a heavy arched girder between the two rims of the gear. The two different sections of the gear shown in this Figure are in planes displaced 22½ degrees against the other. Before the gear housing is shown the pedestal supporting the after bearing of the turbine and securely fastened to the gear housing.

Figure 16 shows the method of lubricating the pinions as installed in the reduction gears of the *Bremen*. The lubricating oil is brought to a nozzle manifold into which are inserted individual interchangeable nozzles through which the oil is brought up to the active part of the meshing teeth. One simple motion only is required to exchange one of these nozzles should it become clogged; after loosening of a dog and turning aside a clamp a nozzle with its pipe can be pulled out and replaced by a new one.

The whole installation of lubricating oil piping has been the subject of extraordinary care. A total of about 160 tons of lubricating oil are in circulation, in the sump and gravity tanks for the four turbine units and the pumps as well as filters and coolers are liberally dimensioned.

The four main lubricating oil pumps, one for each of the main propelling units have a capacity of 270 cubic meters (71,400 U. S. gallons) of lubricating oil per hour each and the four reserve lubricating oil pumps of customary duplex type can deliver 90 cubic meters (23,800 U. S. gallons). There are three lubricating oil filters and three coolers of about 110 square meters (1184 sq. ft.) cooling surface each for each main engine room of two main propelling units.

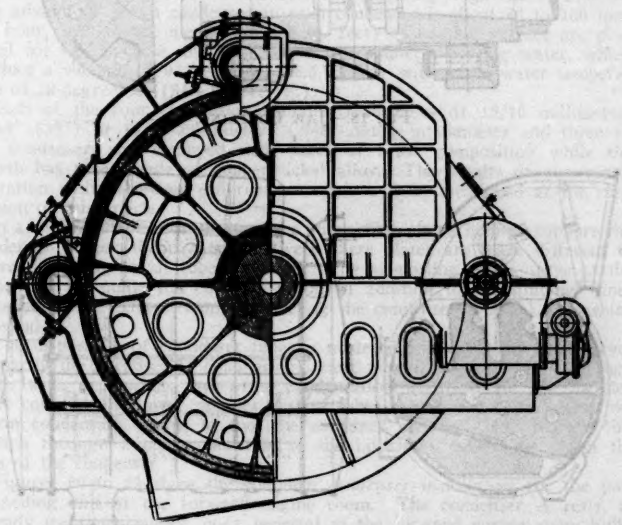


FIG. 17.—GEAR DRIVE. END VIEW.

The capacity of the lubricating oil tanks is also liberally dimensioned, they are installed partly as gravity and partly as sump tanks. A total of four sump tanks of 67 cubic meters (17,700 gallons) combined capacity with six overflow tanks of about equal combined capacity are in existence. In addition to the above mentioned tanks provision is made in the double bottom of the ship to store about 50 cubic meters (1321 gallons) of clean lubricating oil and about 114 cubic meters (3000 gallons) of dirty or used oil.

Besides these supply tanks arranged in the double bottoms four gravity tanks of 20 cubic meters (528 gallons) each arranged at the height of the boat deck in the engine room shafts serve to supply the lubricating oil for all requirements.

The used lubricating oil from the turbines and gear housing drains into the corresponding sump tank, next to which is an overflow tank, which receives the surplus oil from the sump tank. Only when the lubricating oil pump stops and the oil runs completely out of the gravity tanks will the overflow tank be filled as well as the regular sump tank. If overflow tank were not provided the lubricating oil level would rise in the gear housing.

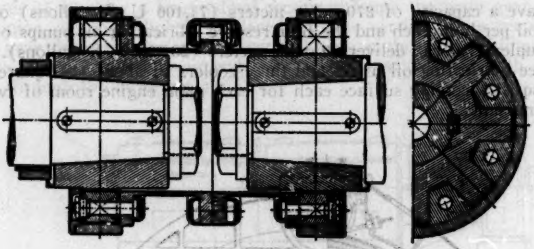


FIG. 18.—JAW COUPLING.

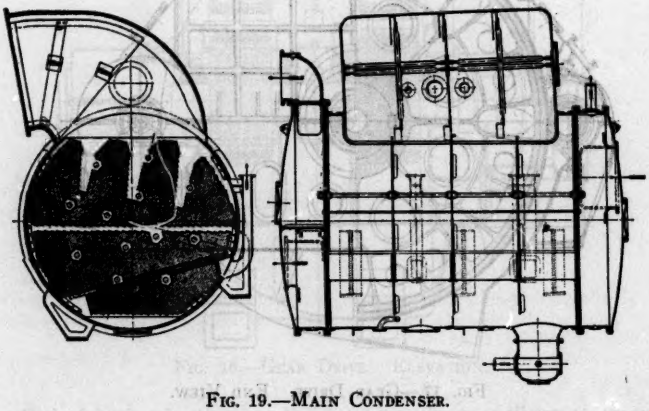


FIG. 19.—MAIN CONDENSER.

The lubricating oil pumps are arranged next to their respective sump tanks from where they suck the oil and deliver it under pressure into the main lubricating oil piping which leads through the filter and cooler into the gravity tank; from here the oil is distributed by gravity to the individual lubricating places.

The reciprocating oil pumps are piped in parallel with the main lubricating oil pumps and serve not only as stand-by units but also as transfer pumps, and for this purpose they are connected to all supply, overflow and dirty oil tanks, so that they may even be used to balance the amount of oil in the systems of all the propelling units.

A motor-driven lubricating oil purifier is installed for each propelling unit, which operates continuously, taking a small amount of oil out of the main lubricating oil piping and returning it cleaned back into the sump.

The above-mentioned main lubricating oil pumps are direct turbine driven and of vertical type, installed under the floor directly aft of the gear drive housing. The delivery pressure of these pumps was set at 60 meters (197 feet) to make doubly sure that the height to the lubricating oil gravity tank will be overcome with ease. The capacity of these as already mentioned is 270 cubic meters (71,400 gallons) per hour and requires an 80-100 B.H.P. turbine to drive them. This turbine which in addition to governor regulation has also a simple hand throttle, is fed from auxiliary superheated steam pipe system with steam of 23 atmospheres (327 lbs. per sq. in.) pressure and works non-condensing with a steam consumption of about 23 kilograms (50.8 lbs.) per B.H.P. hour; its speed is about 3200 R.P.M.

With regard to the main condensers the following might be of interest. The amount of steam condensed in each condenser is about 80 to 100 tons per hour, 1850 square meters (20,000 sq. feet) of cooling surface are provided for this purpose and 75 times that amount in cooling water, which produce a vacuum of 95 per cent (28.5 inches) with a sea water temperature of 18 degrees C. (64.4 degrees F.).

Each of the four condensers contains 7850 tubes of 19/16 millimeters (.748"-.630") or 19/15 millimeters (.748"-.591") in diameter and three of the condensers are equipped with tubes of brass composition while the fourth has tubes made of copper-nickel alloy. The results of one year's operation with these two different materials will be mentioned at the conclusion of this article.

In addition to the tube sheets three supporting plates help to support the condenser plates; the distances between these plates are made different to prevent building up of tube vibration. The supporting plates serve at the same time to guide the entering steam in addition to the guiding vanes installed in the exhaust trunk connecting the condenser to the L.P. turbine housing.

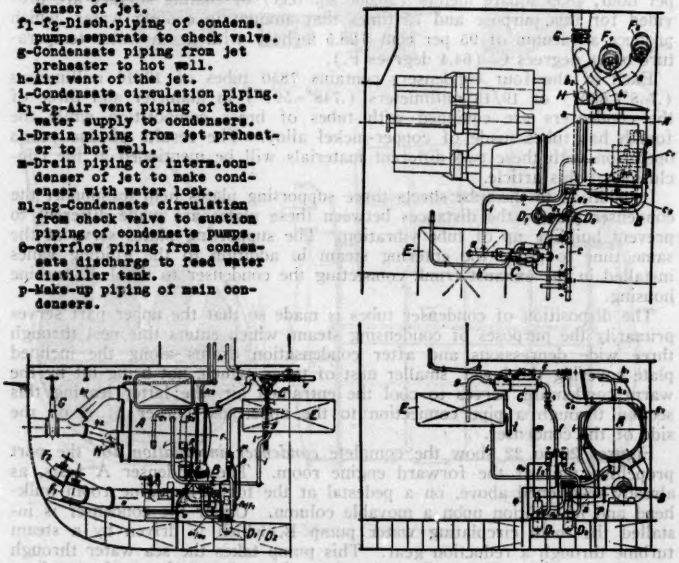
The disposition of condenser tubes is made so that the upper part serves primarily the purposes of condensing steam, which enters this nest through three wide depressions and after condensation drains along the inclined plate covering the lower, smaller nest of tubes, which, not being hit by the warm condensate, serves to cool the entrained air, the latter leaving this section through a pipe connection to the uppermost corner of it on the side of the condenser.

Figures 20 to 22 show the complete *condenser installation* for the port propelling unit of the forward engine room. The condenser A rests, as already mentioned above, on a pedestal at the forward engine room bulkhead and in addition upon a movable column. Under the condenser is installed the main circulating water pump B, which is driven by a steam turbine through a reduction gear. This pump takes the sea water through the two sea valves F_1 and F_2 and delivers it to the condenser from where it is again expelled outboard through the pipes b_1 and b_2 and the stop valve G.

The condensate flows out of the condensate pot on the condenser through the pipe e to the two condensate pumps D_1 and D_2 which force it as a cooling medium through the inter cooler C of the steam jet end into the hot well E. The condensate pumps of each engine room are interconnected by means of pipe f.

LEGEND TO FIGS. 20-22

A-Main Condenser.
B-Main circ.water pump.
C-Steam jet.
D1-D2-Condensate pumps.
E-Hot well.
F1-F2- Sea valves of the main circ.water pumps.
G-Shut off gate valve of the main circ.water pump.
H-Regulating butterfly valve of the circ.water drain piping.
a-Suction to main circ.water pump.
b1-b2-Circ.water disch.from main condenser outboard.
c-Air suction of steam jets.
c1-Separate air suction from condenser.
c2-Combined air suction piping.
e-Combined suction piping of condensate pumps.
e1-e2-Suction piping of cond. pumps.
f-Combined discharge piping of condes.pumps to interm.condenser of jet.
f1-f2-Disch.piping of condens. pumps,separate to check valve.
g-Condensate piping from jet preheater to hot well.
h-Air vent of the jet.
i-Condensate circulation piping.
k1-k2-Air vent piping of the water supply to condensers.
l-Drain piping from jet preheater to hot well.
m-Drain piping of interm.condenser of jet to make condenser with water lock.
n1-n2-Condensate circulation from check valve to suction piping of condensate pumps.
o-overflow piping from condensate discharge to feed water distiller tank.
p-Make-up piping of main condensers.



FIGS. 20-22.—CONDENSER INSTALLATION.

The air is sucked out of the condenser through both pipes c_1 and c_2 by means of a two-stage steam jet which is mounted directly above the already mentioned intercooler C. The condensate pump piping and the suction piping c are interconnected with the respective piping systems of the starboard propelling unit, so that one steam jet, which is large enough to handle double the amount of air, can still maintain the proper vacuum for both turbine units, should the other steam jet fail to operate.

The total *boiler installation* of the passenger liner *Bremen* consists of 20 water tube boilers, 11 of which are of the double ended type, while 9 are single ended. All of these boilers are arranged into four firerooms and are oil-fired.

In order to secure the best possible utilization of available space and out of consideration for the best service and assembling conditions these boilers were installed as follows:

Boiler room I—2 double ended and 3 single ended.
II—3 double ended.
III—3 double ended and 3 single ended.
IV—3 double ended and 3 single ended.

The following are the principal dimensions of the double and single ended boilers:

Heating surface of one double ended boiler—1100 sq. m. (11,840 sq. ft.).
Heating surface of one single ended boiler—550 sq. m. (5920 sq. ft.).
Total heating surface of all 20 boilers—17,050 sq. m. (183,500 sq. ft.).
Heating surface of one superheater—62.5 sq. m. (672 sq. ft.).
Heating surface of superheaters for one double ended boiler—250 sq. m. (2690 sq. ft.).
Heating surface of superheaters for one single ended boiler—125 sq. m. (1345 sq. ft.).
Total heating surface of all superheaters—3500 sq. m. (37,200 sq. ft.).
Heating surface of air preheaters for one double ended boiler—568 sq. m. (6110 sq. ft.).
Heating surface of air preheaters for one single ended boiler—282 sq. m. (3035 sq. ft.).
Total heating surface of all air preheaters—8786 sq. m. (94,600 sq. ft.).
Water space of one double ended boiler—24.74 cub. m. (6540 gals.).
Steam space of one double ended boiler—8.94 cub. m. (2360 gals.).
Water space of one single ended boiler—13.81 cub. m. (3670 gals.).
Steam space of one single ended boiler—5.81 cub. m. (1535 gals.).
Steam pressure—23 atmos. (327 lbs. per sq. in.).
Steam temperature—375 degrees C. (707 degrees F.).

Each boiler consists of one upper drum and two lower drums, which are connected to the upper by means of two tube nests. The upper drum has an internal diameter of 1700 millimeters (66.93 ins.) and the lower drums a diameter of 850 millimeters (33.47 ins.). Both upper and lower drums are of riveted construction, their heads are of elliptical section with a large radius of curvature and they are also riveted, see Figures 23-24.

The drums themselves, their heads and steam domes are made of open hearth steel, having a tensile strength of 41 to 48 ng/mm^2 (58,000-68,000 lbs. per sq. in.) and an elongation of 20 to 25 per cent.

The double ended boilers are provided with two steam domes, while the single ended boilers have only one, the flange of the steam domes being riveted to the body of the drum. The water tubes are expanded in the tube sheets and the tube holes are provided with an internal groove.

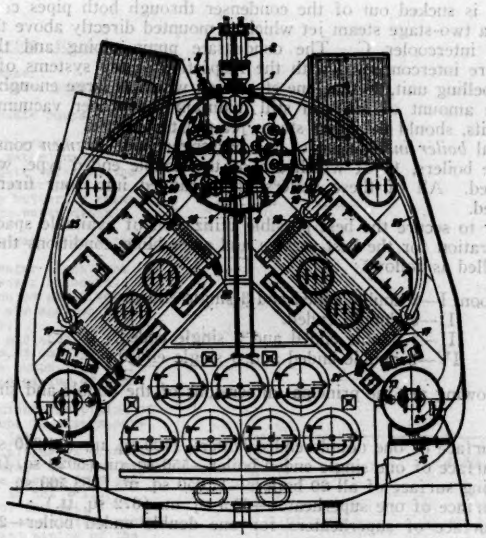


FIG. 23.—BOILER FRONT SHOWING FITTINGS.

The tube nests on each side of the boiler consist of two groups and these are so formed that a space exists between them for the purpose of locating superheaters there.

The flue gases pass through the superheater from the top downward and through the boiler tube nests from the bottom upward. Each superheater consists of 14 bent tubes (seamless steel tubing), a saturated steam header and a superheated steam header. Each double ended boiler has four such superheaters and each single ended boiler two, with the exception of boilers for port use which have no superheaters fitted. The superheaters may be easily withdrawn from the boiler by means of a special trolley arrangement for inspection purposes. The superheater tubes are so supported in a frame made of heat-resisting (calorized) sheet steel that they can be pulled out individually out of the boiler housing.

The superheater headers are outside of the boiler housing fastened to the sheet steel shell of it. Steam and flue gases flow in counterflow through the superheater and therefore the saturated steam header is at the bottom while the superheated steam header is above. The superheater headers are made of open hearth forgings and fitted with connections for draining, superheat steam connections for soot blowing, etc.

Boilers and their housings are covered with mats of spun glass in asbestos casings.

Air preheaters consist of steel tubes which are expanded in forged steel tube sheets and all tubes are equipped with retarders.

Air preheaters extend over the whole length of the boiler housing. In the double ended boilers the air passes the preheaters twice crosswise, while in the single ended boilers it passes them only once and in longitudinal direction.

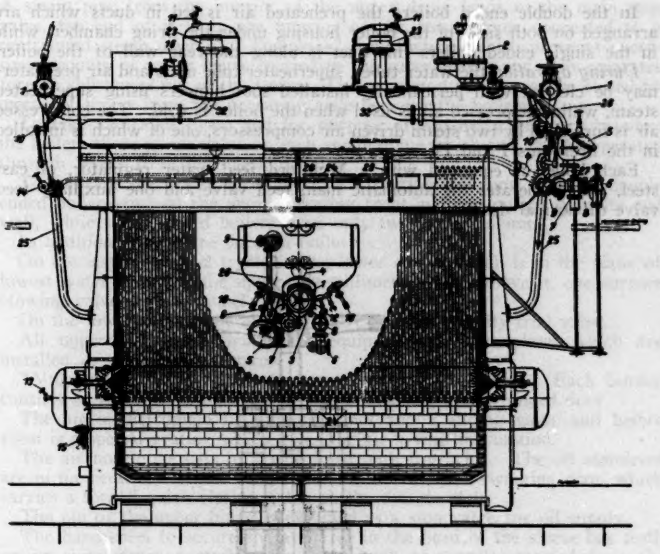


FIG. 24.—DOUBLE ENDED BOILER AND ITS FITTINGS.

LEGEND TO FIGS.23-24.

1-Main stop valves.
2-Quadruple High-Lift Safety Valve (cockburn)
3-Spherical fitting for feed valves.
4-Y-piece for feed pipes (in upper drum).
5-Mumford regulator (float tank).
6-Mumford feed valve.
7-Combined aux.feed valve (check and stop valve).
8-Stop valve for Mumford feed valve.
9-Stop valves for Mumford regulator.
10-Water column.
11-Gage valves (vent valves).
12-Try cocks.
13-Salinity try cocks.
14-Surface blow valve.
15-Bottom blow valve.
16-Pressure equalization valve -stop valve for saturated steam.
17-Superheater header for wet steam.
18-Superheater header for superheated steam.
19-Outlet on superheater header.
20-Superheater safety valves.
21-Drain valves on wet steam. header of superheater.
22-Soot blower valves on superheat end of superheater header.
23-Centrifugal steam drier.
24-Zinc boxes.
25-Connection bet.main stop valve and superheater header.
26-Connection bet. Mumford float tank and boiler drum.
27-Conn.bet.Mumford float tank and boiler bottom.
28-Distribution pipe for feed water.
29-Surface blow piping.
30-Low water alarm.
31- " " " (high rate).
32-Steam lead-off pipe.

In the double ended boilers the preheated air is led in ducts which are arranged on both sides of the boiler housing under the firing chamber, while in the single ended boilers this duct is along the rear wall of the boiler.

During operation the water tubes, superheater tube nests and air preheaters may be cleaned with permanently installed soot blowers using superheated steam, while compressed air is used when the boiler is cold. The compressed air is furnished by two steam driven air compressors, one of which is installed in the firerooms I and IV.

Each boiler is equipped with a Mumford feed water regulator, of cast steel, which operates the automatic main feed valve and one auxiliary feed valve of normal design.

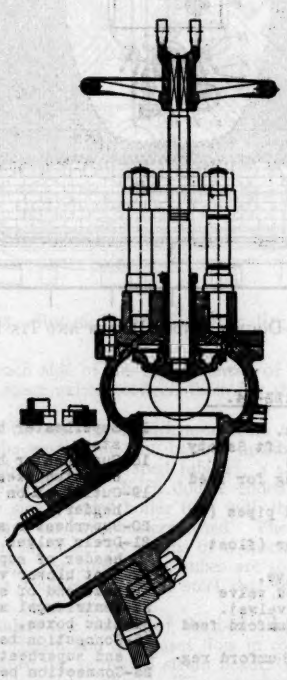


FIG. 25.—MAIN STEAM VALVE ON BOILERS.

Each double ended boiler is equipped with a quadruple and each single ended a double high lift safety valve of "Cockburn" design, installed on the body of the upper drum.

Furthermore the double ended boilers received two and the single ended boilers one main stop valve (Figure 25) of 140 millimeters (5.51") internal diameter from which the steam flows to the superheater headers of the respective boiler. The housing, head and valve disk are made of nickel.

A steam pipe from the interior of the upper drum leads to the main stop valve, the end of it is in the interior of the steam dome connected to a centrifugal steam drier of "Tompkins" manufacture, which prevents excessive amounts of water being carried to the superheaters and also separates foreign substances out of the steam. The main stop valves may be closed by means of levers from the operating aisle between the boilers.

To take saturated steam and for purpose of equalizing pressure between the boilers there is one valve on each steam dome which can also be operated through levers from the operating aisle.

To indicate the water level there are 3 water columns on each double ended boiler—two on the main operating front and one on the opposite end wall, while single ended boilers have only two water columns.

In additional there are on each boiler:

On the upper drum—2 trycocks, the lower one of which is in the plane of lowest water level and the upper 100 millimeters (4") above it, one surface blowing valve with a funnel;

On the lower drum—one blow-off valve and one salinity trial valve.

All upper and lower drums are equipped with zinc plates which are installed in suitable steel boxes.

"Vulcan" oil firing system was used to fire the boilers. Each burner consists of a frame fastened to the boiler housing and a hinged door.

The air is distributed by guiding vanes and a sliding valve and before them is a special grating which gives the air a whirling motion.

The air nozzle consists of a ring lined with fire brick. The oil atomizers are in the centrally located tubular hub or sleeve of the swinging door, which carries a forged quick closing valve on the outside of it.

The pin of the upper hinge is designed as a stop valve for oil supply.

The handwheel to secure the atomizer in the head of the sleeve has teeth on its circumference, meshing with the teeth of a rack operated from the quick closing valve. This interlocking device prevents loosening of the atomizer in the sleeve before the quick closing valve is closed (patent by P. Müller).

Each boiler front has seven burners, so that the total number of burners on all boilers amounts to 217.

Before the manufacture of the boilers was started the builders decided to construct one double ended boiler and subject it to extensive steaming experiments. In addition to the determination of the boiler efficiency various extensive measurements were made in the interior of the boiler to determine the temperatures of combustion, of the flue gases and of air, also pressure drops in the passes and canals, as well as heat transfer in the different parts of the heating surface.

A 24-hour test made on the final design of the boiler gave the following excellent results: With a feed water temperature of 36 degrees C. (96.8 degrees F.) 32,700 kilograms (72,200 lbs.) of water were evaporated with the expenditure of 2720 kilograms (6000 lbs.) of fuel oil, or 29.7 kilograms per square meter of the heating surface (.205 lbs. per sq. foot) or 12.03 times the amount of water as compared with oil burned. The average steam temperature was 360 degrees C. (680 degrees F.); the pressure—20.5 atmospheres (291 lbs. per sq. in.). The low pressure of steam is due to the fact that the design pressure was raised from 21 atmospheres to 23 atmospheres only after the boilers were manufactured, in order to compensate effectively for the pressure drop expected from the long piping.

CO_2 measurements gave for the flue gases beyond the air preheater a CO_2 content of 13.67 per cent which corresponds to a 4 per cent excess of air; the average boiler efficiency as determined by this experiment was 88.1 per cent, which is an excellent result.

The construction of flue gas passes and the temperature distribution in them is shown schematically on Figure 26. The calculated combustion temperature of the oil was about 2160 degrees C. (3920 degrees F.), before the entrance into the first tube nest a temperature of 1000 degrees C. (1832 degrees F.) was determined by means of Seger cones; the rest of the temperatures shown in the Figure 26 were determined by calibrated thermocouples. The per cent figures written in the tube nests give the ratio of the heating surface of the group considered referred to the total wetted heating surface, which was taken equal to 100 per cent.

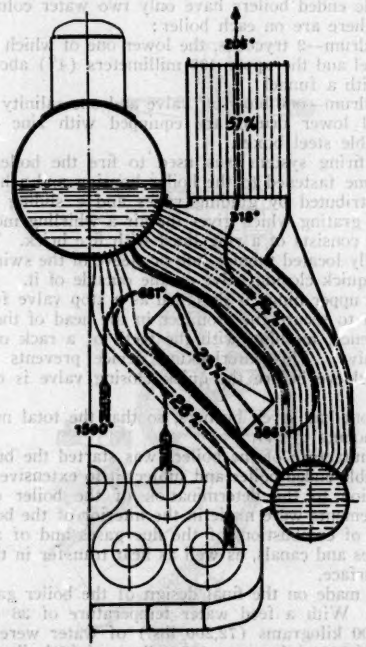


FIG. 26.—TEMPERATURE DISTRIBUTION IN BOILER.

The results of calculation, based on measured temperatures, assuming variable specific heat of the flue gases, as to heat absorbed by individual parts of the boiler are presented in the form of a heat balance, shown on Figure 27. The diagram shows that a very large portion of the heat introduced with the fuel oil—72 per cent—are already absorbed by the first water tube nest of the boiler, which has in round figures 26 per cent of the total heating surface. The radiation from the combustion chamber to the first tube rows is responsible therefore for the major part of the heat transfer.

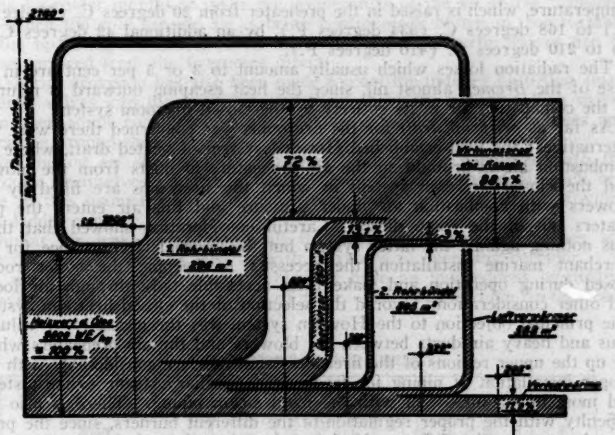


FIG. 27.—HEAT BALANCE OF A DOUBLE ENDED BOILER.

Theoretische Verbrenungstemperatur = theoretical combustion temperature.
Hiezwert d. Öles = Heating value of oil.
Rohrbündel = Tube nest.
Wirkungsgrad des Kessels = Efficiency of boiler.
Uberhitzer = Superheater.
Luftvorwärmer = Air preheater.
Verlustwärme = Waste heat.

The superheater following the first nest of tubes which utilizes the temperature range between 681 degrees C. (1258 degrees F.) and 386 degrees C. (727 degrees F.) according to calculations is responsible for an additional 13.1 per cent of the heat contained in the fuel oil. Next follows the second tube nest, which the gases enter with a temperature of 386 degrees C. (727 degrees F.) and leave with a temperature of 318 degrees C. (604 degrees F.). Due to small temperature difference between the flue gases and the water in the tubes the average heat transfer of this nest is naturally very small compared with its heating surface. It absorbs only 3 per cent of the heating value of the fuel oil from the flue gases—apparently a very unsatisfactory result, which, however, cannot be changed, since the cooling down of the flue gases to a lowest possible temperature requires naturally very large heat exchanging surfaces.

The sum of the above three parts of heat absorption gives 88.1 per cent, which corresponds to the already mentioned efficiency of the boiler. The warm flue gases carry with them 11.9 per cent of the heat introduced with the fuel oil and this constitutes the stack losses.

The liberally dimensioned air preheater following the second tube nest cools down the flue gases to 206 degrees C. and absorbs from them 4.1 per cent of the heating value of fuel. This heat is utilized by increasing the combustion temperature and thereby the useful temperature drop, although this is increased also by the already mentioned circumstance, that the combustion air on its way to the fuel burners passes around the outer skin of the boiler. The latter feature is responsible for raising the combustion air

temperature, which is raised in the preheater from 20 degrees C. (68 degrees F.) to 168 degrees C. (334 degrees F.), by an additional 42 degrees C. or up to 210 degrees C. (410 degrees F.).

The radiation losses which usually amount to 3 or 5 per cent are in the case of the *Bremen* almost nil, since the heat escaping outward is returned to the combustion air in its entirety by the closed fireroom system.

As far as *artificial draft* for the firerooms was concerned there were two alternatives: the well-established Howden system of heated draft, where the combustion air is brought to the air preheaters in ducts from the blower, and the closed fireroom system in which the firerooms are filled by the blowers with combustion air under pressure and this air enters the preheaters out of the room direct. Careful consideration showed that there was nothing against the latter system but the unusual circumstance for the merchant marine installation, the necessity of keeping the boiler rooms closed during operation and make them accessible only through air locks. All other considerations favored the selection of the closed fireroom system. The principal objection to the Howden system was the necessity of voluminous and heavy air ducts between the blowers and the air preheaters which fill up the upper regions of the firerooms and interfere considerably with the proper installation of piping for feed water, auxiliary steam, exhaust steam and most important of all with the main steam piping. There is also the difficulty with the proper regulation of the different burners, since the pressure drops are quite different in the various branches of the air ducts and therefore each burner requires its individual regulation of air supply. Besides separate blowers are required to ventilate the firerooms and change the air there as required.

All these three objections are non-existent when the closed fireroom system is used. Ducts for handling air are almost completely eliminated and only short pieces of pipe are necessary to properly distribute the air from the blowers throughout the fireroom. Even air pressure distribution in the fireroom makes the fires even with hardly any special regulation. Change of air in the fireroom is very intensive since all of the combustion air passes through this room.

The decision in favor of the closed fireroom which was based on the above advantages of this system, was much helped by the fact that shortly before that time the original Howden system of the passenger liners *Majestic* and *Berengaria* was taken out and replaced by the closed fireroom system.

Only three single ended boilers, one in each of the firerooms I (port), III and IV (stbd.) received also a special system for operation in port, by means of a special motor driven blower similar to the Howden system. When a butterfly valve is opened on the air preheater and it is possible to go over to using pressure in the fireroom as for all other boilers.

Each fireroom has two blowers driven by turbines through reduction gears, and so dimensioned that each is sufficient to furnish all the combustion air required by the whole fireroom. As foundation for these blowers serve the tops of the fuel oil bunkers, which are arranged on the sides of each fireroom.

The blowers suck the air through liberally dimensioned suction air shafts which follow the uptakes from the sun-deck into the firerooms. Each fireroom is served by two ventilators on the sun deck next to the stacks, which are turned in the direction of ship movement and due to high speed of the latter unload considerably the blowers.

The air suction shafts in their vertical part are not immediately in contact with the uptakes but are separated from them by a special ring-shaped shaft; by opening butterfly valves in the upper part of the shaft and in the bottom covering of it in the fireroom a part of the combustion air can be made to pass in direct contact with the walls of the uptakes; this is used

during he colder months to make the temperature of the air in the fireroom, into which the cold outside air is being continuously blown in, comfortable for the operating personnel. The above-mentioned butterfly valves are of course serviced from the firerooms direct.

A special provision has also been made to cool the uptakes by fitting small butterfly valves into the casing around them above the boilers, which valves when open admit cold fireroom air into the space between the uptakes and the casing around them.

The suction air shafts of the fireroom blowers are connected with the firerooms themselves through a double door or the so-called air lock and serve in this manner also as an emergency exit, the shafts being provided with ladder rungs.

Similar air locks interconnect all the firerooms among themselves and with the engine rooms as well as with entrances to operating corridor.

It must be added also that the blowers have an independent lubricating oil system with filters and coolers of their own. The turbine driving each

TABLE I. NEWEST PASSENGER LINERS.

	NAME	MAURETANIA	BERENGARIA ex-IMPERATOR	LEVIATHAN ex-VATERLAND	MAJESTIC ex-BISMARCK	ILE DE FRANCE	BREMEN
SHIP'S HULL	Year of completion	1907	1912	1914	1921	1926	1929
	Length bet.perpend.-m.	232	270	276	280	233	271
	Molded breadth - m.	26.8	29.9	30.5	30.5	28.0	31.0
	Draft - m.	10.2	11.3	11.7	11.5	9.75	9.75
	Displacement - tons.	38,000	57,000	60,000	64,000	42,000	52,000
	B.R.T.	30,700	52,200	60,000	56,600	43,500	49,900
	Contract speed - knots	25.0	22.5	22.5	23.0	23.5	26.25
MACHINERY	Designed H.P.	68,000	61,000	61,000	65,000	52,000	90,000
	RPM Turbines	180	180	170	180	220	1,800
	RPM Propellers	180	180	170	180	220	182.5
	BOILERS Number	25	46	46	48	20	20
	BOILERS Type	Scotch	W.Tube	W.tube	W.Tube	Capus	W.Tube
	BOILERS Heating Surface - m^2	14,780	18,900	19,550	20,400	13,140	17,050
	BOILERS Grate Area - - m^2	378	350	358	011	011	011
	STEAM Pressure - at.	13.7	16.0	17.9	17.5	16.5	23.0
	STEAM Temperature - °C.	Sat.	Sat.	Sat.	Sat.	Sat.	360
TURBINES	No. of rows of blades.	184	162+3C	224	-	-	57+2C
	L.P. drum dia.- mm	3,560	3,650	3,900	-	-	1,550
	Longest blade - mm	560	600	615	-	-	330
	No. of blades - total	600,000	750,000	817,000	-	-	202,000

blower is designed up to a maximum output of 200 B.H.P., but delivers normally only 70 B.H.P. at 3000 R.P.M.; it consists of an impulse wheel with a double row of blades, having a mean blade diameter of 900 millimeters (35.43").

The turbine speed is reduced by means of a reduction gear to about 750 R.P.M., at which speed the blowers deliver 40 cubic meters (1410 cubic feet) per second. The blower rotors are of the "Turbon" design and are fastened in pairs to the rotor shaft; the outside diameter of such wheels in the forward fireroom is 920 millimeters (36.22") and in the after firerooms they are of 1020 millimeters (40.158") diameter.

TABLE 2. AVERAGE SPEED OF "BREMEN" DURING FIRST YEAR OF OPERATION.

TRIP No.	DIRECTION	DATE	DURATION OF TRIP d. h. m.	DISTANCE COVERED Miles	SPEED - KNOTS OUT	SPEED - KNOTS HOME	SPEED - KNOTS MEAN	REMARKS
		1929						duced - fog.
1	Out	7/18 - 7/22	4-17-42	3164	27.55	-	-	Moderate sea; short time re-
	Home	7/27-8/1	4-14-30	3084	-	27.91	27.87	Light to rough sea.
2	Out	8/16 - 8/20	4 - -	-	27.41	-	-	
	Home	8/23 - 8/28	4-19-48	3183	-	27.51	27.46	Light sea; wind 2-4. -→airplane landing at sea.
3	Out	9/6 - 9/10	4-18-48	3081	26.84	-	-	Continuous fog; delay due to -
	Home	9/13 - 9/18	5 - -	3113	-	25.84	26.30	2 days rough sea with wind 7.
4	Out	9/26 - 10/1	4-17-42	3093	27.19	-	-	Moderate sea - partly fog.
	Home	10/5 -10/10	4-18-30	3119	-	27.24	27.22	Fair; moderate sea.
5	Out	10/18-10/22	4-17-24	3084	27.2	-	-	3 days rough sea and swell.
	Home	10/25-10/30	4-21-12	3117	-	26.7	26.95	Rough sea and high swell; 5 hours reduced speed.
6	Out	11/8 -11/12	4-19-12	3090	26.82	-	-	Continuously rough sea.
	Home	11/15-11/20	4-22 -	3124	-	26.5	26.66	2 days storm; wind up to 11; high seas;long swell.
7	Out	12/6 -12/12	7-7-54	3107	17.6	-	-	Long heavy [illegible] wind 11;
	Home	12/15-12/20	4-19-06	3116	-	27.1	22.35	Moderate to rough sea; wind 6.
		1930						
8	Out	1/5 - 1/9	5-2-42	3092	25.2	-	-	Rough sea with high swell; 2 days reduced speed.
	Home	1/11 - 1/15	4-19-36	3114	-	27.0	26.1	3 days rolling sea and long swell
9	Out	1/24 - 1/28	5-0-42	3092	25.62	-	-	Rough sea; 2 days reduced speed.
	Home	2/1 - 2/5	4-18- -	3114	-	27.3	26.46	Moderate sea; wind 4.
10	Out	2/14 - 2/18	4-22-48	3108	26.2	-	-	Rolling to rough sea; 10 hours reduced speed.
	Home	2/22 - 2/26	4-22-06	3163	-	27.0	26.6	Moderate to rough sea; wind 3-6.
11	Out	3/7 - 3/11	4-21-12	3142	26.8	-	-	Moderate to rough sea; wind 4-8.
	Home	3/15 - 3/19	4-21-48	3162	-	27.03	26.91	Rolling sea, swell; wind 5-7.
12	Out	4/4 - 4/8	5-11-24	3167	24.1	-	-	Rough to high sea; wind 7-11; 20 hours reduced speed.
	Home	4/12 - 4/16	4-20-12	3178	-	27.4	25.75	Moderate sea; wind 2-6.
13	Out	4/25 - 4/29	4-22-04	3151	26.7	-	-	High sea; 6 hours reduced speed due to fog.
	Home	5/3 - 5/7	4-21 -	3199	-	27.3	27.0	Rough sea and swell; rain and light fog.
14	Out	5/15 - 5/19	5-1-36	3166	26.0	-	-	Moderate to rough sea; 8 hours reduced speed due to fog.
	Home	5/21 - 5/25	4-20-54	3195	-	27.3	26.65	Light sea; wind 2-6.
15	Out	6/2 - 6/6	4-19-36	3157	27.3	-	-	Moderate sea; wind 4.
	Home	6/11 - 6/16	4-19-54	3195	-	27.6	27.45	Light sea; 2 hours at anchor, 7 hours reduced speed due to fog.
16	Out	6/21 - 6/25	4-22-42	3155	26.6	-	-	Moderate sea; 9 hours reduced
	Home	6/28 - 7/3	4-20-36	3194	-	27.4	27.0	speed due to fog. Moderate sea and swell; wind 4.

Yearly Average 26.55

During normal operation both blowers in each fireroom are in operation and therefore their regulation, which is by means of a valve in the steam supply pipe is made in common for both; this valve can be operated from the fireroom floor by means of linkage and a liberal number of draft gauges is installed to guide such regulation.

It was desired to keep the wind resistance of the *Bremen* to as low a figure as possible, therefore not only the superstructure, bridge and all other deck structures in general, but even the stacks were made of a special shape. The cross-section of the stacks is of droplike shape, the width of this airfoil section being 6.2 meters (20.3 ft.) and its length—15 meters (49.2 ft.) The height of the stacks on the other hand was made low in order to give the ship a silhouette in keeping with its speed, which measure received general approbation.

Below the stack the uptake is divided into two separate ducts, an arrangement which was chosen to give the artistic taste of the decorators free rein in designing the social halls.

The average height of the stacks above the centerline of the lower burners is 41 meters (135 ft.), the area of the after stack which receives the gases of 6 double ended and 6 single ended boilers is 33.3 square meters (360 sq. ft.), while the forward stack, serving 5 double ended and 3 single ended boilers, has an area of 24 square meters (270 sq. ft.).

In order to relieve the boiler housings from carrying any part of the weight of the uptakes, these are suspended by means of spring cushioned anchors from suitable points of the deck or bulkheads.

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* Schiffbau, Oct. 1 and 15, 1930.

EDITOR'S NOTE: This article will be concluded in the February issue of the JOURNAL.

ASSOCIATION NOTES.

REAR ADMIRAL GEORGE WASHINGTON BAIRD, U. S. NAVY.

AN APPRECIATION.

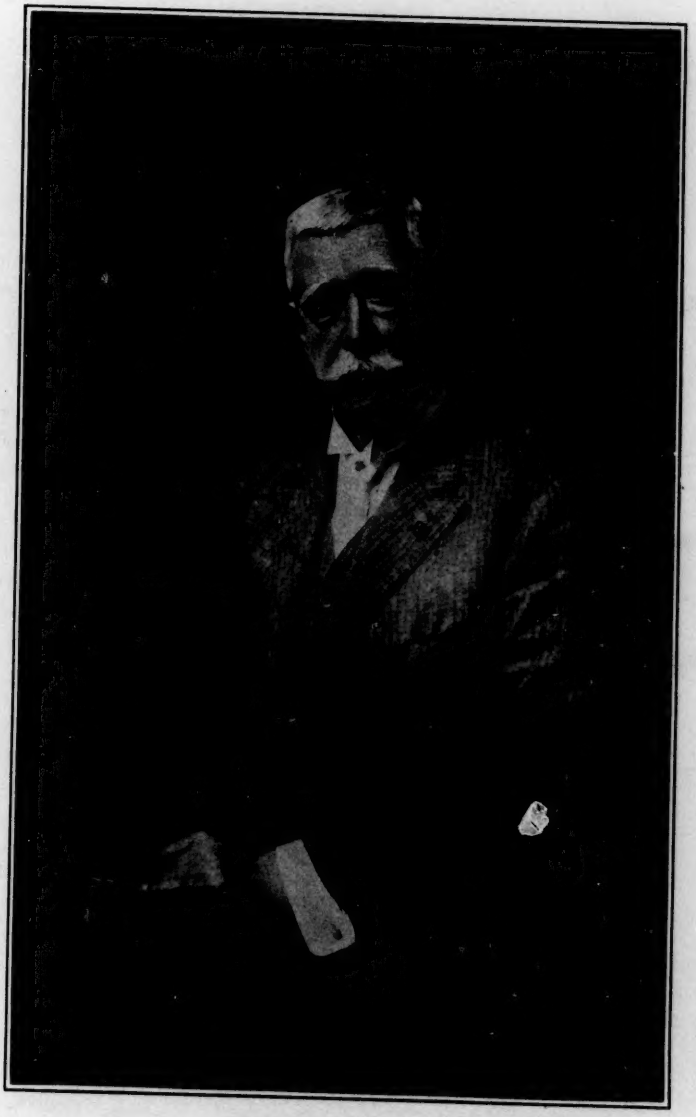
By WALTER M. MCFARLAND, MEMBER.*

The death of Admiral Baird not only marks the end of a long life and distinguished career but also practically the last of the old Engineer Corps as constituted at the close of the Civil War, as he was almost the last survivor of that body of useful and faithful officers, many of whom were men of outstanding ability and all of whom rendered loyal and faithful service in the time of the Republic's greatest need.

It is probably rather difficult for the young officers of our modern navy, where every Line officer is educated as an engineer, and where the splendid facilities of our great engineering school at Annapolis offer every inducement for professional equipment and skill, to realize that there ever was a time when engineering in the Navy held a very subordinate position, and when, indeed, it was an unwelcome although necessary auxiliary.

The old Engineer Corps covered this period of storm and stress. Some of us who came later, and, although graduates of Annapolis, went through the mild form of the obsolescent antipathy, can appreciate the fine morale of the men who performed efficiently the vitally necessary work of engineering in the Navy against so much discouragement. The marvel is not that there were so many good men of fine ability who "saw it through" but that they did not all follow the example of the considerable number of more ambitious (or possibly less tenacious) men who left the Service to become distinguished in civil life. The names of Thurston, Leavitt, Cooper, Fisher, Allen, Newton, Selden, Stimers, Van

* Formerly Chief Engineer U. S. Navy.



GEORGE W. BAIRD

Buren and Greene, as well as that of Haswell, the father of the Engineer Corps, at once occur to those familiar with the history.

Of those who made their career in the Service, two stand out above all others, Isherwood, the great experimentalist and Engineer-in-Chief of the Civil War; and Melville, the Arctic hero and upbuilder of the New Navy as Engineer-in-Chief from 1887 to 1903. Besides these there were other men of fine ability, Henderson, Macomb, Brooks, Loring, Robie, Ayres, Dungan, David Smith, Milligan, Towne, Andrade, Harris, Perry, Albert, Magee and J. A. B. Smith, who retired for age, and Kafer, Manning, Jones, Robinson and others who retired earlier. On behalf of those who, in their early days in the Service, looked up to these older men for example, guidance and encouragement, it is a pleasure to pay a tribute of affection to these and many others whose names are not given for lack of space. It is true that, as in any large group of men, there were some who were none too friendly to the youngsters from the Naval Academy and who referred to them scornfully as "kid glove engineers" and other disparaging epithets, but these were in a very small minority and time has softened any asperity which might have been caused in the days of long ago.

A number of the officers mentioned above were instructors at the Naval Academy and of the greatest influence in the organization of the course in engineering, out of which the splendid facilities of today have grown. Some of these men, like Kafer, Jones, Manning, Crawford and others filled the ideal relation of teacher to pupil, and were always affectionately remembered by those of us who had the benefit of their teaching, and with whom the intimacy of the relation is best expressed by the fact that we all called them "Uncle."

The subject of this sketch belonged to the class of these older men who were progressive and forward looking and who were friends and helpers of their younger associates. In any summary of the outstanding men in the old Engineer Corps, Baird would certainly have occupied a place on the list.

He was born in Washington, D. C., April 22, 1843, and died in Washington October 4, 1930, so that he was in his 88th year. The period of his youth was before the days of widespread opportu-

nities for technical education, so that after the common schools and academy his technical experience was gained in the machine shop and by personal study. He came by his aptitude for engineering naturally, as his father and grandfather were both engineers. The Civil War began before he had reached the voting age and he entered the Navy in the Engineer Corps September 19, 1862, before he was 20 years old. During the Civil War he served on the *Mississippi, Calhoun* and *Pensacola* in the west Gulf blockading squadron, and took part in more than twenty engagements. He had great natural engineering aptitude; was a fine draftsman and had attracted the attention of Engineer-in-Chief Isherwood while on duty in the Bureau of Steam Engineering because we find that in 1869 he was ordered to the Mare Island Navy Yard where Isherwood was Chief Engineer of the Yard. It was during this period that Isherwood conducted his famous experiments on screw propellers, which have remained a classic, and undoubtedly Baird was of considerable assistance in the preparation for and conduct of these experiments. After a year of this duty he was ordered to the old *Pensacola,* flagship of the Pacific Fleet, where he put in a full cruise and was then ordered to the Bureau of Steam Engineering where he served for more than three years. This gave him further opportunity for the display of his ability as a draftsman and designer in which capacity he rendered valuable service. In 1877 he was ordered to the *Trenton* and, after her arrival on the European station, he was detached and ordered to the *Vandalia,* where he put in a cruise of about four years, during which time General Grant made his famous trip around the Mediterranean in that vessel. In 1882 we find him ordered to duty with the Fish Commission. During this assignment he supervised the construction of machinery for the Fish Commission steamer *Albatross* and designed most of the special machinery on that vessel, which proved of great value in her special service. It was at this time also that he installed the electric plant in the *Albatross,* which was the first Government vessel of any nation to utilize electric light.

At the conclusion of duty with the Fish Commission he was ordered to special duty as Assistant Superintendent of the State, War and Navy Department Building. While on this service he

installed the electric lighting in the White House, the current being furnished from the generators in the State, War and Navy Department Building. Duty as Chief Engineer of the Presidential yacht *Dolphin* covered the period from 1892 to 1895, when he returned to duty as Superintendent of the State, War and Navy Department Building, which position he continued to occupy until he reached the then retiring age of 62 in 1905. While on this duty he was for a time also Chief Engineer of the Washington Navy Yard. Inasmuch as the work of this Yard was almost entirely the manufacture of ordnance, this additional duty was almost nominal.

Admiral Baird, as already stated, had great mechanical and engineering aptitude, and made a number of inventions, some of which have been used extensively, especially his evaporator and distiller, which for a long time were the standard for the Navy. He invented a pneumatic indicator for showing the direction of turning of the engine shafts. He also gave great attention to the subject of the ventilation of ships, about which he had written at considerable length, and many of his ideas were subsequently utilized in the methods adopted.

Having always been a very active man, it was only natural that he should be ready to devote himself to any public service where it seemed his experience and ability might make him useful. The very next year after his retirement, a change was made in the method of appointing the Board of Education of the District of Columbia. Admiral Baird was one of the first appointees under the new system, where the Board consisted of nine prominent citizens appointed by the Justices of the Supreme Court of the District. He was elected its President and held this position for several years until he relinquished his membership on the Board.

He had passed through the various grades in the Engineer Corps in due course, becoming a Second Assistant Engineer in 1866, a Passed Assistant Engineer in 1874 and Chief Engineer with the relative rank of Lieutenant Commander in 1892. When the Personnel Bill, which amalgamated the Line and the Engineer Corps, took effect March 3, 1899, he became a Commander in the Line, and in 1902 a Captain. Under the provision of law, when

he retired, he was given the rank of Rear Admiral on the retired list, on account of honorable service during the Civil War.

It is most appropriate in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS that mention should be made of the fact that Admiral Baird was one of the most active participants in the organization of the Society. The Admiral had always kept a diary and under date of September 30, 1888, there is this entry:

"We had a meeting, in the Bureau, today, at which there were present Messrs. Bieg, Chambers, Griffin, Norton, Kaemerling, Mattice, Moore, Nauman and myself and some others. I presided and Mr. Griffin was elected Secretary. The purpose of the meeting was to organize, with the object of printing the results of experiments, records of trial trips, breakdowns, etc., of ships of the Navy, and also for the purpose of reading papers and discussing the debatable subjects in naval engineering."

The Admiral goes on to say that he had previously proposed the printing of a Journal, which proposition had been debated informally, without any idea of forming a Society. At the meeting above referred to Passed Assistant Engineer Mattice offered a motion to organize a Society and to call it The American Society of Naval Engineers. This motion was adopted. His notes also record:

"It was agreed, at that meeting, that I should correspond with the 'old engineers' (of whom I was one) and to invite that prince of engineers, Mr. Isherwood, to edit the JOURNAL. Mr. Griffin who was one of the younger engineers (graduates of the Naval Academy) was to correspond with the other graduates. We were to solicit subscriptions. Our efforts were crowned with success; indeed we builded wiser than we knew, for the plan we adopted, a plan which has ever characterized the Corps, was to record facts exactly as we found them, and this made the JOURNAL popular, if not famous."

Later on his notes say that, in the early days of the Engineer Corps, there was little or no literature on marine engineering. The Corps was recruited from the builders of marine machinery, draftsmen, etc., for there was no other field from which to draw. He adds:

"At that time every engineer made his own text books. We were never without our note books and were never neglectful in the matter of making notes or records. It may, therefore, be understood why and how we came to establish this Society and to print our JOURNAL."

He further notes:

"The Society provided an Advisory Council to take charge of the editing, but the Council usually left most of the editing to the Secretary, who has been, generally, the executive officer. I had the honor of being the second President of the Society an honor of which I have always been very proud. What has made the JOURNAL, and the Society, such a success is the accuracy of the records printed and without bias. We have printed the records as we found them; not as we wanted to find them. The immense amount of tabular information, so tiresome to those not intimately interested, is like a gold mine to the man who wants that precise information. The letters which were received from engine builders during the first year of the JOURNAL's publication, were complimentary and encouraging. I do not think it is too much to say that our first Secretary, Assistant Engineer Griffin, is entitled to more credit than anyone else, in securing the first subscribers and advertisers and in his intelligent editing."

The Assistant Engineer Griffin mentioned above is the present distinguished Rear Admiral Robert S. Griffin (retired), the great Engineer-in-Chief of our Navy during the World War.

Besides his membership in our own Society, Admiral Baird was President of the District of Columbia Branch of The Society of the Sons of the American Revolution; a member of The American Society of Engineers and of the Washington Society of Engineers. He was also a member of the Cosmos Club. He was very prominent and distinguished in the Masonic Order, having been Grand Master of the Masons in the District of Columbia in 1896. At the time of his death he was the oldest living Past Grand Master; he had been elected a 33rd degree Mason in 1906.

Admiral Baird's long life covered almost the entire history of marine engineering. He did not indeed go back to the very ear-

liest days, but, when he began, there were still in the Service vessels of the earliest type, and he had the opportunity of personal acquaintance with the very earliest engineers of the Navy, such men as Haswell, Isherwood, Wood, Shock, Macomb and others. It must have been a great satisfaction to him, after passing through the stormy period of his early days, when he felt that engineering was not receiving its proper recognition, to live to the period when, at the suggestion of a distinguished Line officer, the Engineer Corps was absorbed in the Line and the performance of engineering duty became a recognized qualification of every Line officer.

It may not be out of place here to refer to the remarks made by Assistant Secretary (afterwards President) Roosevelt in forwarding the report of the Personnel Board of 1897 which recommended this amalgamation, and where he said substantially that it had come about naturally because in the modern navy every officer had to be an engineer whether he wanted to or not. He said:

"We are not making a revolution; we are merely recognizing and giving shape to an evolution, which has come slowly but surely and naturally, and we propose to reorganize the Navy along the lines indicated by the course of the evolution itself."

He also referred to the striking analogy between what was then proposed for our Navy and the similar combination in the latter part of the 17th century in Great Britain, when the Navy was reorganized on the basis of combining the soldier, who had formerly been only a passenger except in the time of action, with the seaman who provided for the movement and navigation of the ship, the amalgamated officer being the naval officer and the amalgamated enlisted man the man-of-wars-man. In the same way our new naval officer had to be a "fighting engineer."

It is a melancholy pleasure to have contributed this testimonial to the memory of an old friend who had been kind to him in his early days in the Service, and it seemed a suitable time to bear testimony to his position as an outstanding figure in the old Engineer Corps and to pay a tribute to the memory of the men who constituted that Corps. It is a great satisfaction to the writer that, as a member of the Personnel Board of 1897, he had a part in the scheme for the reorganization of the Navy along the lines

from which the present development has occurred. While his own naval career terminated by resignation within a few months after this great change took effect, he has always maintained a keen interest in the progress of the Navy; and it is a great pleasure to bear testimony to the high efficiency of its personnel and to the fact that as a body of men the officers of the American Navy have no superiors.

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With keen regret the Society records the death of REAR ADMIRAL CHARLES W. DYSON on November the eighth. The limited time available before publication has made it impossible to give an appreciation of Admiral Dyson's life and work until the next number of the JOURNAL, when a tribute prepared by Captain W. M. McFarland will be published.

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At the regular Annual Meeting of the Society held October 7, 1930, the following were nominated for officers for 1931:

For President:

Rear Admiral H. E. Yarnell, U. S. N.

For Secretary-Treasurer:

Captain O. L. Cox, U. S. N.
Commander Henry F. D. Davis, U. S. N.

For Member of Council:

Rear Admiral C. W. Dyson, U. S. N.
Rear Admiral John Halligan, U. S. N.
Captain I. E. Bass, U. S. N.
Captain Henry Williams (CC), U. S. N.
Captain H. S. Howard (CC), U. S. N.
Commander H. M. Cooley, U. S. N.
Commander C. A. Dunn, U. S. N.
Commander Bryson Bruce, U. S. N.
Commander Jno. Q. Walton, U. S. C. G.
Mr. J. F. Metten.
Mr. H. M. Southgate.
Mr. W. M. McFarland.
Mr. C. P. Wetherbee.
Mr. B. P. Lamberton.

Ballots are now outstanding and will be counted on December 26, 1930.

The Society voted to hold a banquet in 1931 in accordance with its custom to have one of these events annually.

MEMBERSHIP.

The following have joined the Society since the publication of the August, 1930, JOURNAL:

NAVAL.

Clark, W. B., Lieutenant, U. S. N. R., 5 Park St., Peabody, Mass.

Gelhaus, Henry F., Lieutenant, U. S. N. R., 615 18th Avenue, San Francisco, Calif.

Johnson, Warren W., Lieutenant, U. S. N.

Smith, Roy C., Commander, U. S. N.

Sypher, H. H., Jr., Lieutenant Marine Corps Reserve, Bethlehem Steel Co., Bethlehem, Pa.

Walton, Fred W., Lieutenant, U. S. N.

CIVIL.

Jansson, A. H., Marine Review, Penton Building, Cleveland, Ohio.

ASSOCIATE.

Bonner, James B., 603 Wilkins Building, Washington, D. C.

Byrne, Francis H., 355 Mountain Way, Rutherford, N. J.

Preysler, Col. Carlos, Apartado 815, Madrid, Spain.

Smith, John H., Westinghouse Electric and Manufacturing Co., Lester, Pa.

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